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Steel railway bridges of large span,

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Figs. 1 to 15, pp. 88 to 97.

A design was wanted for a railway bridge of about 60 m. (197 feet) span to carry five tracks.

The space available being restricted both as regards width and height, the construction of two separate bridges, one for two tracks and the other for three tracks, had to be faced.

The bridge was to be built across the river Ourthe near the station at Chênée, between an existing reinforced concrete road bridge and the factory buildings of the Vieille-Montagne Company. The plan (fig. 1) shows that the space between the road bridge and the factory was only just enough to get in the two bridges, and therefore it has been necessary to adopt a girder section as narrow as possible. The section (fig. 2) which is that adopted for the centre panels of the top boom, and the section (fig. 3) of the centre panels of the lower boom, show the difficulty of the problem.

Moreover, it was necessary to leave a clear headway of 4.60 m. (15 feet) above the maximum navigable high water level of the river of 65.10 m. (213.58 feet) above datum. The level of the under side of the bridge must therefore be not lower than 69.70 m. (228.68 feet). The rail level is 72.22 m. (236.94 feet) and there is therefore only 72.22 m. — 69.70 = 2.52 m. (8.26 feet) between the rail level and the under side of the bridge structure. As there are points and crossings on the bridge, the rails should be laid on ballast, for which the depth of 2.52 m. (8.26 feet) is only just sufficient.

Main girders of simple N lattice work would have had at least ten panels, each about 6 m. (19 ft. 8 in.) long. In view of the great length of these panels (6 m.) and of the distance between the main girders, the cross girders would have had to be extremely large. In the two bridges, and especially in that carrying the three tracks, it would not have been

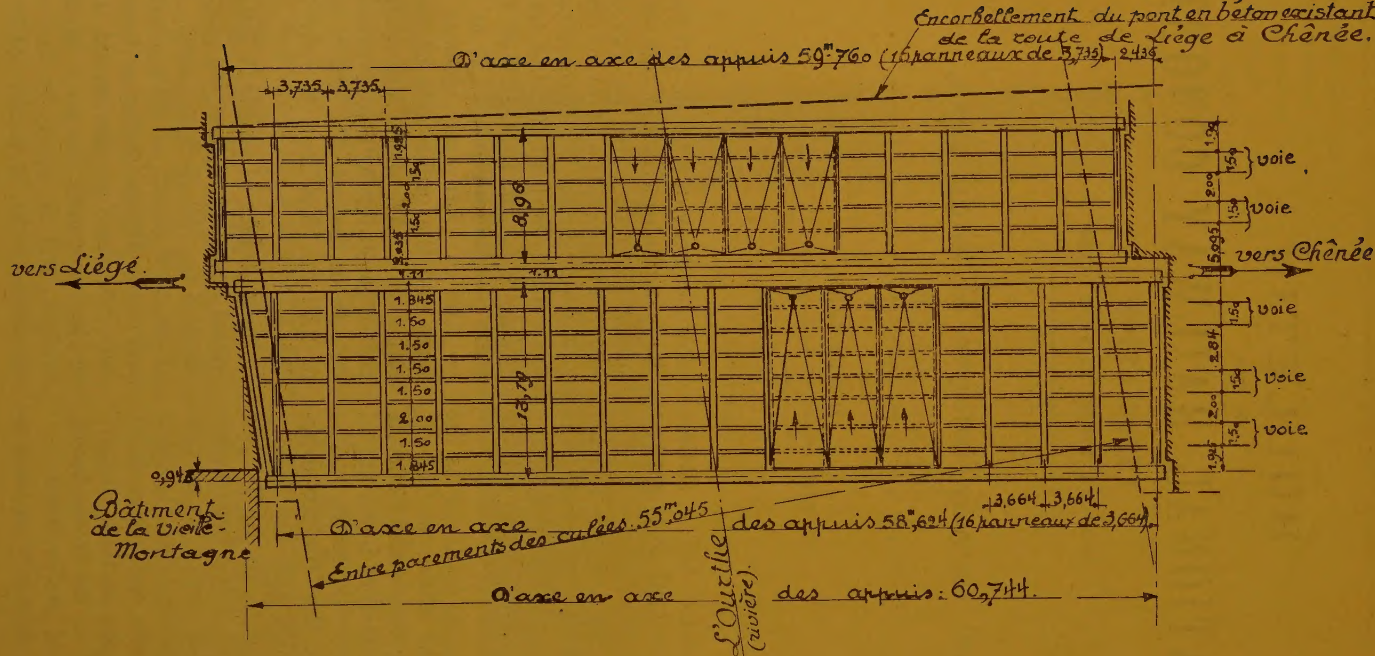


Fig. 1. — Plan — Scale : 1 in 500.

Explanation of French terms ; D'axe en axe des appuis 59 m. 760 (16 panneaux de 3.735 = 196.064 feet centre to centre of supports (16 panels each 12.254 feet). — Encorbellement du pont en béton existant de la route de Liége à Chênée = Parapet of existing concrete bridge carrying the road from Liege to Chênée. — Vers Liége = To Liege. — Vers Chênée = To Chênée. — Voie = Track. — Bâtiment de la Vieille-Montagne = Vieille-Montagne building. — Entre parements des culées 55 m. 045 = Between abutment facings 180.593 feet.

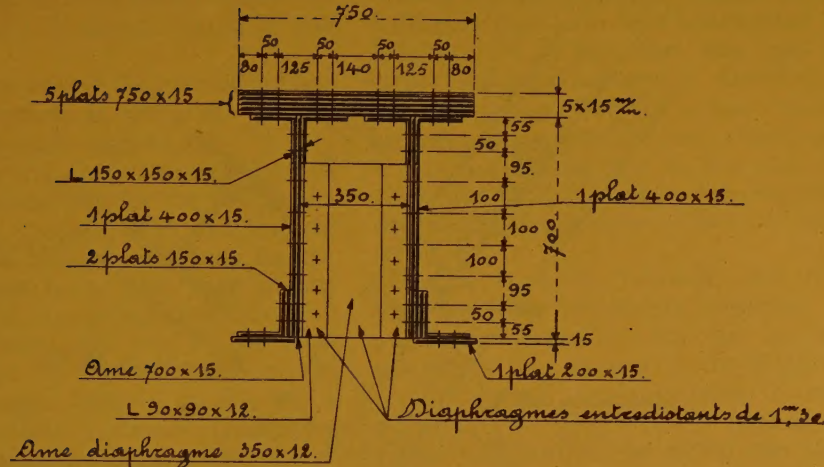


Fig. 2.

Explanation of French terms : 5 plats 750 × 15 = 5 plates 750 mm. × 15 mm. (2.952 × 0.590 inches). — Ame 700 × 15 = Web 700 mm. × 15 mm. (2.755 × 0.590 inches). — Diaphragmes entredistants de 1 m. 30 = Stiffeners 1.30 m. (4.265 feet) apart. — Ame diaphragme 350 × 12 = Stiffening web 350 mm. × 12 mm. (13.780 × 0.472 inches).

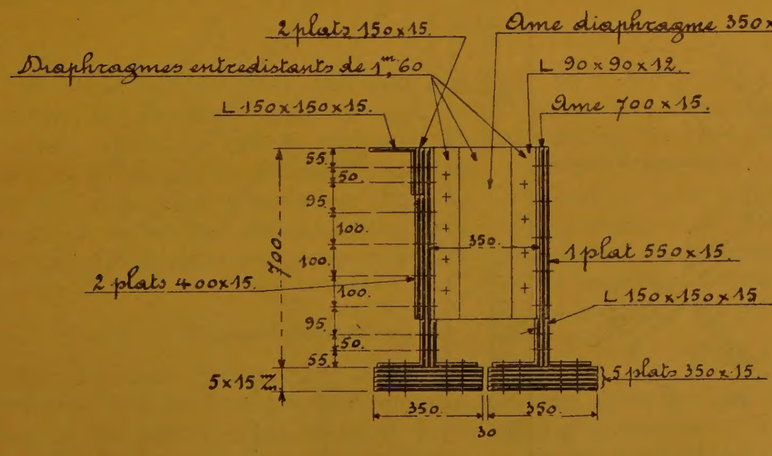


Fig. 3.

possible to have made the members sufficiently strong, on account of the lack of available height. Moreover, simple trellis work would have required very strong longitudinal members.

It was therefore necessary, both from a practical and an economical point of view, to abandon the simple trellis girder and adopt another system.

We have chosen the Baltimore type of

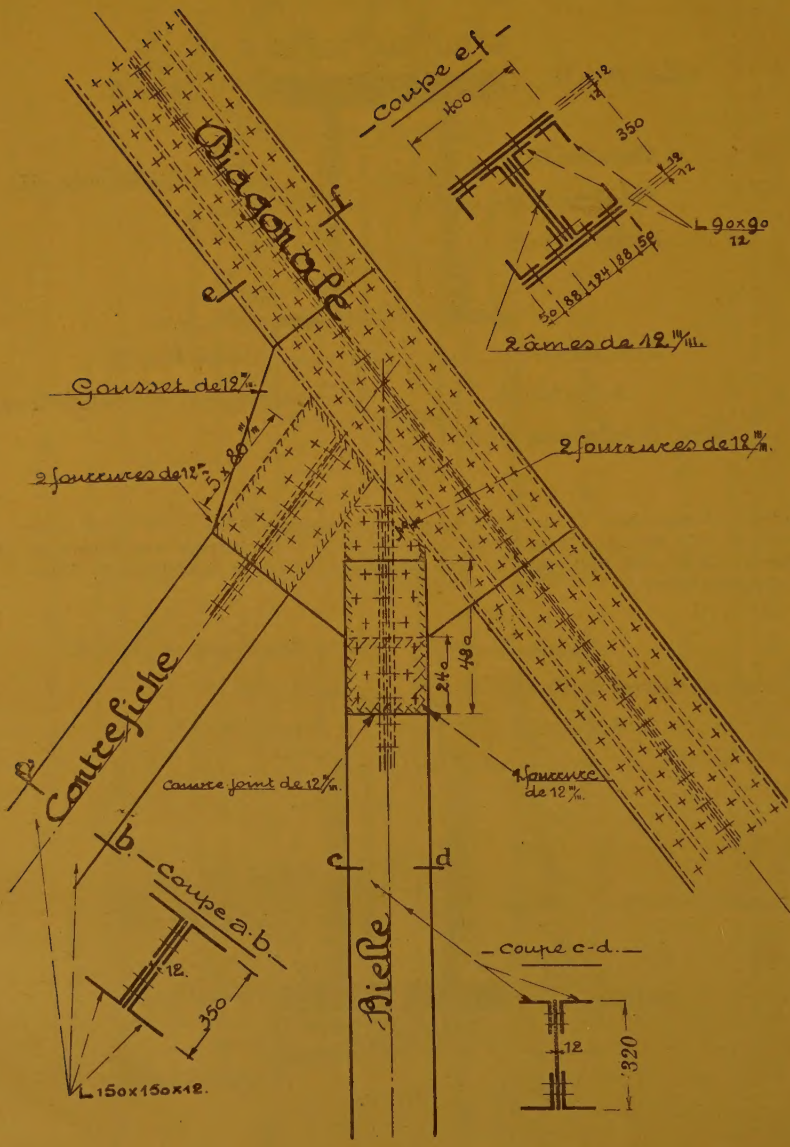


Fig. 4.

Explanation of French terms : Coupe ef = Section ef. — 2 âmes de 12 mm. = 2 web plates 12 mm. (0.472 inch) thick. — 2 fourrures de 12 mm. = 2 packing pieces 12 mm. thick. — Gousset de 12 mm. = Gusset plate 12 mm. thick. — Contrefiche = Strut. — Couvre joint = Cover plate. — Bielle = Tie.

girder, which is a relatively simple lattice work of a N shape with a tie and strut.

This design halved the span of the longitudinal members as well as the distance between the cross girders.

The latter are, in this case, of reasonable dimensions even for the bridge carrying three tracks. The longitudinal members can also be very much lighter.

The bridge being only very slightly on the skew, the bridge work is built squarely, except the end of the three track bridge at the abutment at the Vieille-Montagne factory. In this way it has been possible to provide effective top wind bracing with rigid supports at each end of the bridge.

We believe that up to the present, no railway bridges with main girders of the Baltimore type have been constructed in Belgium. It is true that this type of girder, although favoured in America, is not often suitable for this country where bridges of long span are rare. Moreover, it has been found that in this class of girder the joint formed by the diagonal, the tie and the strut, is difficult to construct.

Figure 4 shows, however, that this joint has been very simply arranged.

It is considered by some that the calculation of the stresses in the elements of this system of triangulation is complicated. We give below a method of general application by which the tensions in the various bars of the lattice work can be determined quickly.

INFLUENCE LINES GIVING FORCES IN THE MEMBERS IN THE BALTIMORE GIRDER.

In this investigation we shall take into account the indirect action of external forces. We know that in this case the influence line may be deduced from that drawn under the assumption that the action is direct by considering the force polygon at the point where the members meet.

1. — Upper boom.

For the stress in the upper boom, we draw the influence lines giving the moment due to the action of an external force P = 1 t. acting indirectly upon the girder, this moment being taken at the point where the girder can be considered as separated in two portions along the line II′ to obtain the stress in the member which we wish to consider.

The line representing these moments is then the same as for an ordinary N girder. We have, for example, in figure 5 the influence line for moments at the point D as affecting the member C′D′. It is only necessary to consider the girder as divided into two portions and to divide the sum of the moments due to a certain condition of loading by μ the perpendicular distance of the member CD from the point about which moments are taken.

2. — Lower boom.

Here again we draw the influence lines giving the moments at a point about which moments are to be taken to determine the force in the member in question. This will be at C′ for the member CD which we will consider as an example (fig. 6).

We see at once that from A to C and from D to K we have the same considerations as for an ordinary N girder. Let us consider the value of the moment when

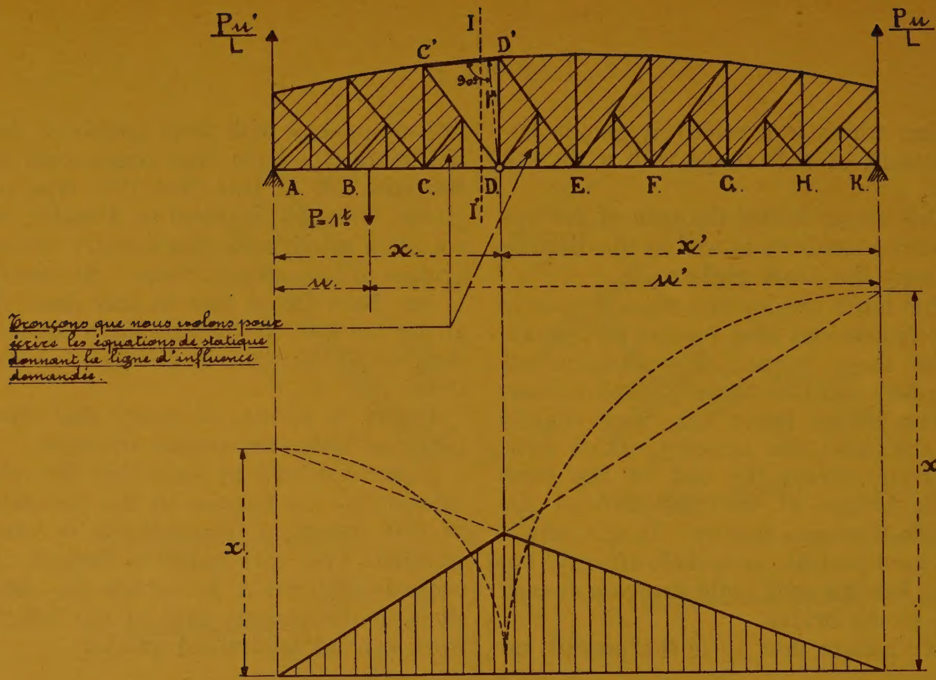


Fig. 5.

Explanation of French terms : Tronçons que nous isolons pour ecrire etc. = Section which is considered as isolated in writing the statical equation to give the required influence line.

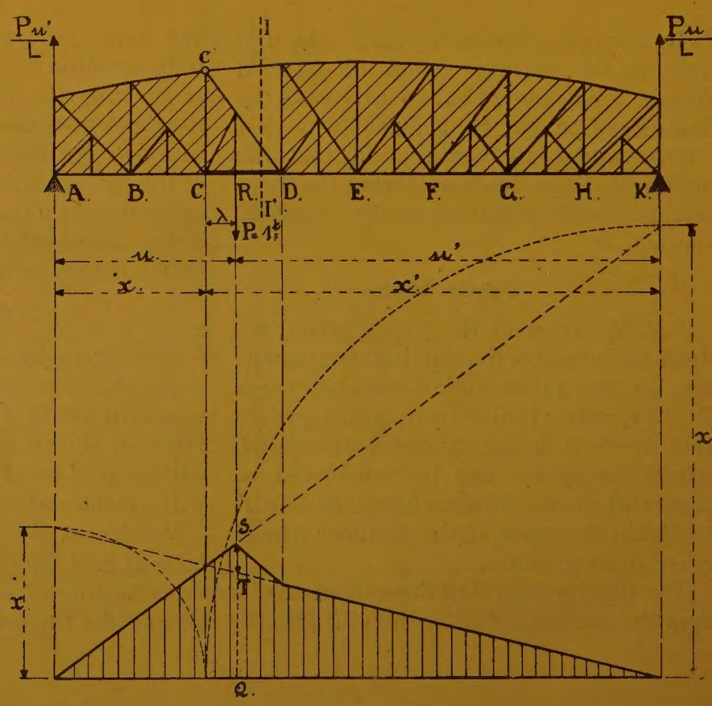


Fig. 6.

the moving force P = 1 t. is at R. It may be noted that we are dealing with a general case which is equally applicable when CR = RD.

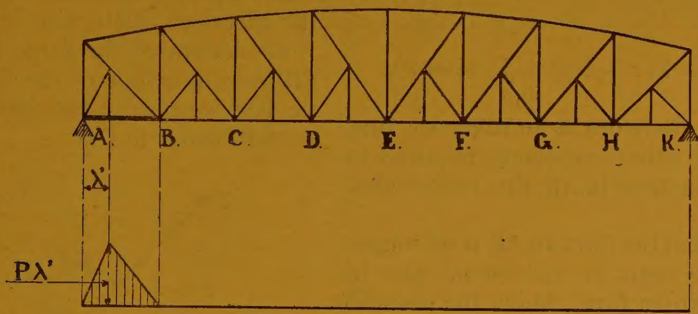


Fig. 7.

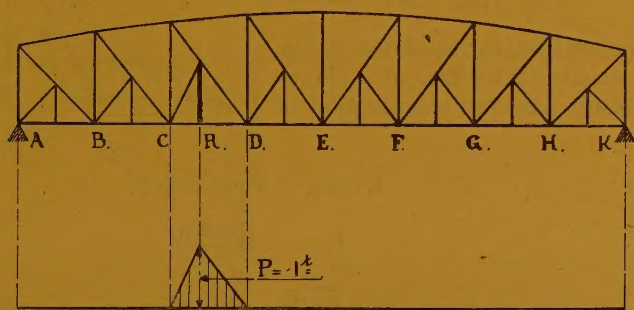


Fig. 8.

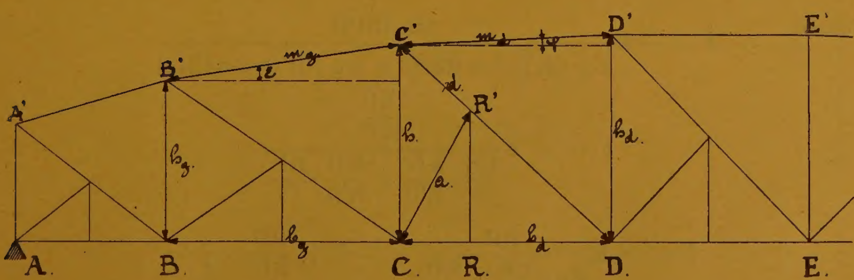


Fig. 9.

We know that the moment at C' is equal to $-\frac{Pu}{L}x'$, that is to say, it is the same value as if the law obtaining from A to C should still hold good. Between C and R and between R and D we have a linear law due to the indirect action of exterior forces.

Let us consider the value of ST.

We have ST = SQ — TQ.

$$= \frac{Pu}{L} x' - \frac{Pu'}{L} x =$$

$$= \frac{P}{L} [(x + \lambda) x' - (x' - \lambda) x] = P\lambda.$$

We have therefore as an influence line necessary to give the moment required to calculate the stress in AB, that represented by figure 7.

We see that the force in AB is no longer theoretically zero, as will be the case in the same girder from which the ties and struts have been removed.

3 — Ties.

The influence lines for the forces in the ties are so simple that sufficient explanation will result from an examination of figure 8.

4. — Struts.

We will first give in figure 9 the notation employed to designate the length of the members dealt with in the formulæ.

To determine the force F in the strut CR′ as a function of the force E in the tie RR′ we resolve the forces along XX′ perpendicular to C′D

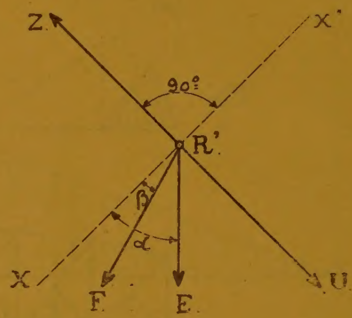


Fig. 10.

We have :

$$E \cos \alpha + F \cos \beta = 0.$$

Whence

$$F = - E \frac{\cos \alpha}{\cos \beta}.$$

But

$$\cos \alpha = \sin \widehat{RR'D}$$

$$\cos \beta = \sin \widehat{CR'D} = \sin (\widehat{CR'R} + \widehat{RR'D}).$$

Therefore

$$F = - E \frac{\sin \widehat{RR'D}}{\sin \widehat{CR'R} \cos \widehat{RR'D} + \sin \widehat{RR'D} \cos \widehat{CR'R}}$$

$$F = - E \frac{\frac{RD}{R'D}}{\frac{CR}{a} \cdot \frac{RR'}{R'D} + \frac{RD}{R'D} \cdot \frac{RR'}{a}}$$

or :

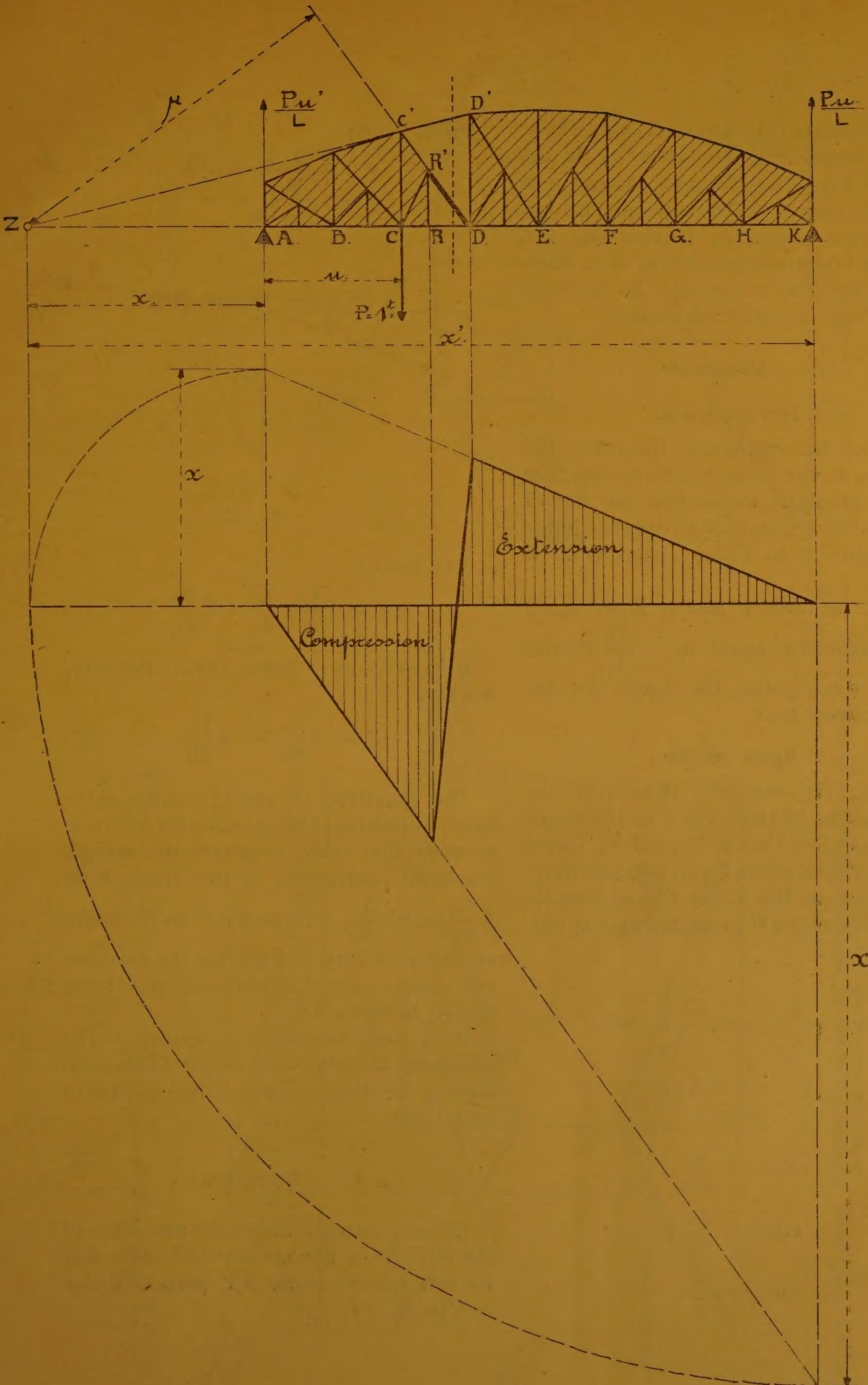
$$F = - E \frac{RD}{RR' \left(\frac{CR}{a} + \frac{RD}{a} \right)} = - E \times \frac{RD}{RR'} \times \frac{a}{b_d}.$$

But

$$\frac{RD}{RR'} = \frac{b_d}{h}.$$

From which

$$F = - E \frac{a}{h} \quad \text{. . . . .} \quad (1)$$



Z
$\frac{Pu'}{L}$
$\frac{Pu}{L}$
C'
D'
R'
μ
A.
B.
C
R
D.
E.
F.
G.
H.
K.
u.
P=1t.
x.
x'.
x
Extension
Compression
x'

Fig. 11.

This formula (1), which is absolutely general, shows that the forces in the tie and strut have always opposite signs; also that in the case of the through span bridge the ties are always in tension and the struts in compression; while in a deck span bridge the ties would be in compression and the struts in tension.

5. — Diagonals.

1. Lower portion.

Drawing the influence lines for the moments at the point Z for the member R′D, for example, we see that the line of moments for a simple N girder still holds good from A to C and from D to K (fig. 11).

When the force P acts at R we again have a moment equal to $-\frac{Pu}{L}x'$ and between these points the linear law due to the indirect load.

2° Upper portion.

Let us, for example, determine the influence line of the force Z in the member C′R′ as a function of X_g and X_d, forces in the portions of the boom adjacent to C′. Let us isolate the point C′ and resolve the forces along YY′ perpendicular to CC′. (See figure 12.)

We have :

$$Z \sin \gamma + X_d \cos \varphi = X_g \cos \varepsilon,$$

from which

$$Z = \frac{X_g \cos \varepsilon - X_d \cos \varphi}{\sin \gamma}.$$

But

$$\cos \varepsilon = \frac{b_g}{m_g}.$$

$$\cos \varphi = \frac{b_d}{m_d}.$$

$$\sin \gamma = \frac{b_d}{d}.$$

From which

$$Z = \frac{X_g \frac{b_g}{m_g} - X_d \frac{b_d}{m_d}}{\frac{b_d}{d}}.$$

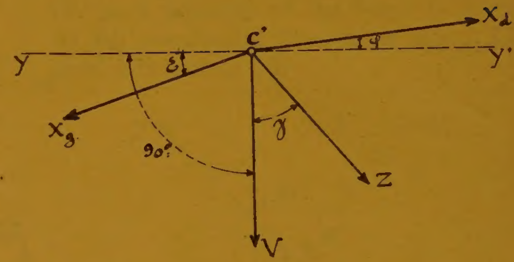


Fig. 12.

And finally

$$Z = d \frac{X_g}{m_g} \cdot \frac{b_g}{b_d} - d \frac{X_d}{m_d} \quad . \quad . \quad (2)$$

If $b_g = b_d$, as is nearly always the case, we have

$$Z = d \frac{X_g}{m_g} - d \frac{X_d}{m_d} \quad . \quad . \quad . \quad (2')$$

The equations (2) and (2′) allow us to easily obtain the influence line for forces Z, because this only requires the simple algebraic subtraction of two linear laws $d\frac{X_g}{m_g}$ and $d\frac{X_d}{m_d}$. These allow us to obtain an influence line which is for the member C′R′ under consideration of the form shown in figure 13.

It is only necessary to calculate the ordinates at either end of the panel containing the diagonal which one wishes to investigate.

6. — Uprights.

Let us consider the influence line of the force V in the upright CC′ For this we can resolve along NN′ perpendicular to C′D (fig. 14).

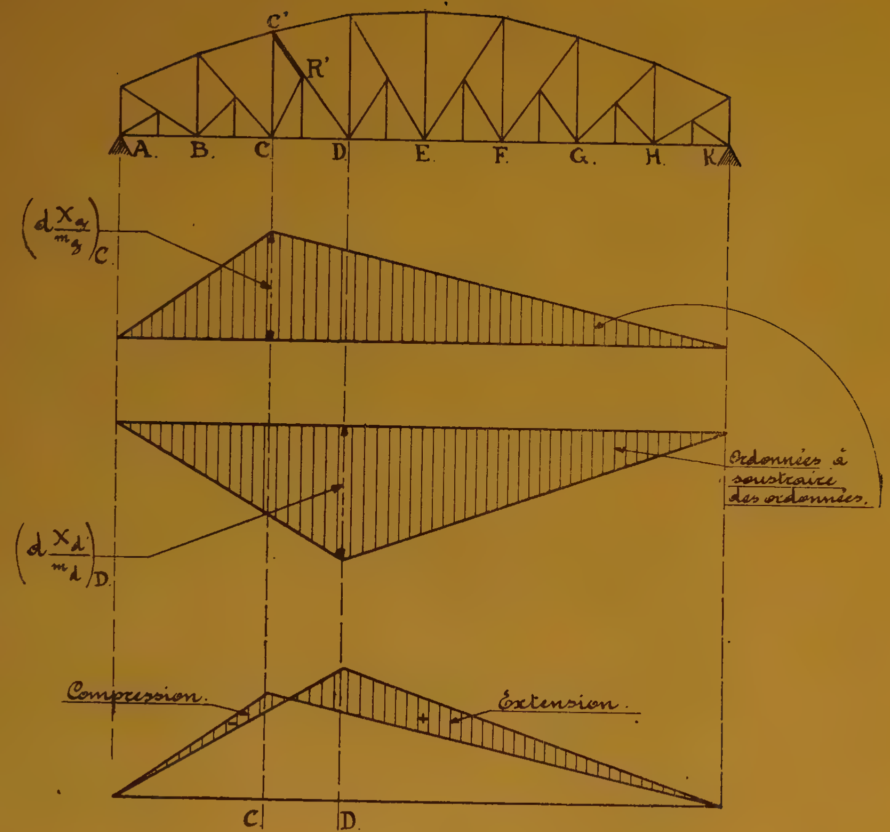


Fig. 13.

Explanation of French terms : Ordonnées à soustraire des ordonnées = Ordinates (see direction of arrow) to be deducted from ordinates (ditto).

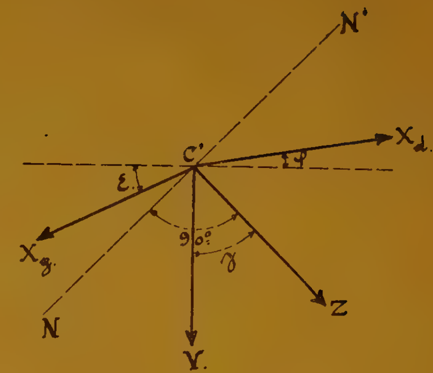


Fig. 14.

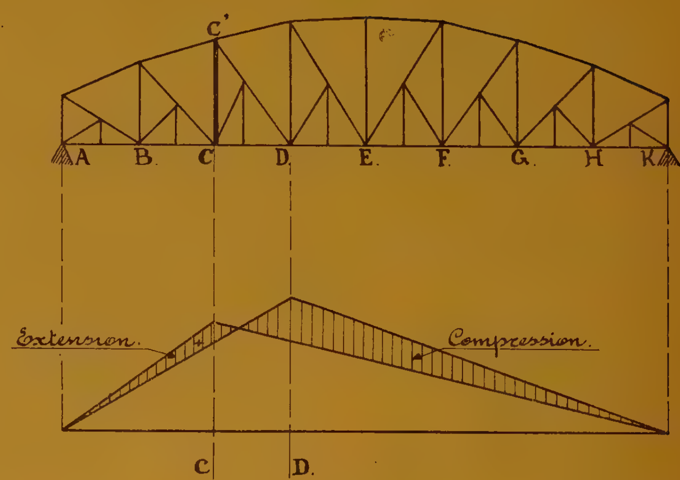


Fig. 15.

We have :

$$V \sin\gamma + X_g \sin(\varepsilon + 90^\circ - \gamma) = X_d \sin(\varphi + 90^\circ - \gamma).$$

Whence

$$V = \frac{X_d \cos(\varphi - \gamma) - X_g \cos(\varepsilon - \gamma)}{\sin\gamma}.$$

Or :

$$V = \frac{X_d\,[\cos\varphi\cos\gamma + \sin\varphi\sin\gamma] - X_g\,[\cos\varepsilon\cos\gamma + \sin\varepsilon\sin\gamma]}{\sin\gamma}$$

But noting that :

$$\sin\varepsilon = \frac{h - h_g}{m_g}. \qquad \cos\varepsilon = \frac{b_g}{m_g}.$$

$$\sin\varphi = \frac{h_d - h}{m_d}. \qquad \cos\varphi = \frac{b_d}{m_d}.$$

$$\sin\gamma = \frac{b_d}{d}. \qquad \cos\gamma = \frac{h}{d},$$

we can write :

$$V = h_d\frac{X_d}{m_d} - \left[h\left(1 + \frac{b_g}{b_d}\right) - h_g\right]\frac{X_g}{m_g} \quad (3)$$

$_g = b_d$ we have :

$$V = h_d\frac{X_d}{m_d} - (2\,h - h_g)\frac{X_g}{m_g} \quad (3')$$

Equations (3) and (3′) allow us to easily draw the influence lines for the forces in the uprights. We have therefore the influence line for the member CC′ as shown in figure 15.

The calculation of the ordinates at the extremities of the panel through which it is necessary to consider the girder as being divided is similar to the calculation for an upright n the case of an ordinary N beam (that is to say, without ties or struts), provided that we have the necessary influence lines.

It may be added that the formulæ given (1) (2) (3) are absolutely general, that is to say, they are applicable to panels of an unequal length (skew bridges for example) for girders carrying through span or deck span bridges, whatever the form of the top boom which becomes the bottom boom in the case of deck span bridges.

The whole of the foregoing may obviously be also applied to road bridges, but the loading due to heavy locomotives is so great that especially for railway bridges, the use of the Baltimore girder becomes desirable.

[385. (01 (.6)]

The Trans-African and Trans-Sahara railways,

By A. FOCK.

Figs. 1 to 3, pp. 104 to 106.

(*Revue politique et parlementaire.*)

The « *Trans-African* » — whether the name be applied to the rail and water route which is being developed between the Cape and Cairo, or to an unbroken railway route between the Cape and Algiers, — will in any case make use of the great trunk line, which now connects Cape Town to Bukama and even to Stanleyville.

This last mentioned centre, which is at the commencement of the great bend of the Congo, and at the junction of the trade routes to the North-East and North-West, happens to be approximately half way between the Southern extremity of Africa and the Mediterranean ports.

The Trans-African railway is therefore half completed, and the work of constructing the Northern sections, — the Trans-Sahara with the extension of the Chad branch to the Congo, — will certainly be no heavier than that involved in building and operating the Southern sections.

These sections, generally speaking, are giving very satisfactory results, both from an economic and financial point of view. The conditions under which the Northern sections will be built and operated will differ in many ways, so that the estimated costs and receipts cannot be determined by a direct comparison. This does not reduce the utility, if not the necessity, of taking into account any main principles established by experience gained on the Southern sections, both during construction, and after they were opened for traffic.

It is therefore of value to consider the question from this aspect, and to assist the reader the essential facts are summarised in the following article.

* * *

The railway from the Cape to Bukama, and the mixed rail and water route to Stanleyville, are at present worked by four companies, the first two of which control all the railways of the Union of South Africa and of Rhodesia, whilst the other two are concessionaires of the railways in the Belgian Congo, with the exception of that from Matadi to Kinshasa.

The following table shows the situation at a glance:

1. *South African Railways and Harbours* (1) :

From the Cape (Capetown via De Aar and Kimberley to Vryburg . . . 1 226 km. (762 miles).

2. *Beira and Mashonaland and Rhodesia Railways* (2) :

This administration includes the following companies :

a) *The Rhodesia Railway Company, Limited* :

From Vryburg via Mafeking to Bulawayo . . . . 946 km. (588 miles).

To be carried. . . 946 km. (588 miles). 1 226 km. (762 miles).

N. B. — Notes (1) and (2), are given as appendices at the end of this article.

Carried . . .	946 km. (588 miles).	1 226 km. (762 miles).
From Bulawayo via Victoria Falls to Kalomo.	602 — (374 —).	
b) *The Mashonaland Railway Company, Limited* :		
From Kalomo to Broken Hill . .	452 — (281 —).	
c) *The Rhodesia-Katanga Junction Railway and Mineral Company, Limited* :		
From Broken Hill to Sakania (frontier of the Belgian Congo). . .	212 — (132 —).	
Total a), b), c)		2 212 km. (1 375 miles).
3. *Lower Congo-Katanga Railway Company* (3) :		
From Sakania (Rhodesian frontier) to Bukama		713 km. (443 miles).
4. *Upper Congo to the Great African Lakes Railway Company* (4) :		
From Bukama to Kongolo (river navigation)	640 km. (398 miles).	
From Kongolo to Kindu (railway) . .	355 — (220 —).	
From Kindu to Ponthierville (river navigation)	320 — (199 —).	
From Ponthierville to Stanleyville (railway)	125 — (77 —).	
Total from Bukama to Stanleyville		1 440 km. (894 miles).
Total length from the Cape to Stanleyville . .		5 591 km. (3 474 miles).

Throughout the whole of the Southern portion of the Trans-African railway, the 3 ft. 6 in. gauge predominates. This state of affairs is easily explained.

When railways were introduced in South Africa, the intention was to provide Cape Colony with a small railway system. As the lines passed through very hilly country, it was thought advisable to adopt the English narrow gauge of 3 ft. 6 in. Between 1873 and 1890, the lines were gradually pushed forward to Kimberley and Vryburg.

Since then the foresight of Cecil Rhodes transformed this line, which had been laid to the frontier between Cape Colony and Bechuanaland, into the nucleus of the Trans-African system, the Southern portion of which is now completed. Although the line was gradually extended, each new section was laid to the 3 ft. 6 in. gauge. This gauge was not therefore deliberately chosen for the great South to North trunk line; it was the result of a state of affairs already existing when the idea of the Trans-African railway was first conceived.

* * *

Faced with the unexpected problem of working as a direct inter-colonial line the various sections which had been constructed — a total length of nearly 3 500 km. (2 180 miles) up to the Belgian Congo frontier — the two South African Railway Companies, took steps

N. B. — Notes (3) and (4) are given as appendices at the end of this article.

to make the best use possible of the system they had to operate.

The results obtained are nothing short of remarkable, especially on the South African Railways, on which the methods of operation, which have been very highly developed with great ability, have reduced the disabilities of the narrow gauge to a minimum.

* * *

In the first place it was necessary to strengthen the track so as to carry heavy axle loads at relatively high speeds. By using rails weighing 40 kgr. per metre (80 lb. per yard) and very careful laying and ballasting them the necessary strength has been obtained.

As a result of the work done, the long distances are covered at an average speed of 52 km. (32 miles) per hour, notably in the case of the « Union Express », which runs between the Cape and Johannesburg, a distance of 1 540 km. (960 miles) (1). Shorter runs, as the 400 km. (250 miles) from Witbank — the junction of the Pretoria and Johannesburg lines — to Lorenzo-Marquez are made at an average speed of 60 km. (37 miles) per hour. Such timings are exceptional on 3 ft. 6 in. gauge railways.

* * *

A great effort was required of the Running Department before such trains could be worked. The existing stock includes powerful engines of the *Pacific* (4-6-2), *Mountain* (4-8-2) and *Mallet* (2-6-6-2) types, which in running order weigh 77, 87 and 97 t. respectively — excluding the tenders — and develop a drawbar tractive effort of 13 600, 18 100 and 21 700 kgr. (30 000, 39 900 and 47 800 lb.).

Improvements in construction and a high standard of maintenance have made it possible to increase the length of the runs which engines can make without returning to the shed. This results in faster timings for the trains and fewer locomotives in service, which saves time and money.

The maximum through run by one and the same locomotive is that from Mafeking to Bulawayo and back (1) — more than 1 600 km. (994 miles) — two sets of men, who travel with the train, alternately taking charge of the engine.

* * *

The Beira and Mashonaland and Rhodesia Railways have not yet reached the standard of the South African Railways, but are gradually approaching it, thanks to a capable and progressive management.

The use of rails weighing 30 kgr. (60 lb. per yard made it necessary to strengthen the road which is in progress between Bulawayo and Broken Hill, and at several places additional sleepers had to be laid. A number of bridges which were not strong enough to carry the increasing weight of the trains, and especially of the locomotives had to be strengthened.

In order to meet the ever increasing demands of the traffic, it is proposed to adopt heavy articulated engines, either of the *Mallet* or *Garratt* types. On the 2nd and 3rd sections of the Trans-African line, from Vryburg to Sakania, *Pacific* or *Mountain* types, similar to those used on the South African Railways, will, generally speaking, be sufficient.

In any case, it may be predicted that in a few years time, express trains will be able to run from the Cape to the Belgian

(1) The route is via Kimberley, and thus includes the 1st section of the Trans-African railway.

(1) The section from Vryburg to Bulawayo, although belonging to the Beira and Mashonaland and Rhodesia Railways, is worked by the South African Railways.

Congo at an average speed of 52 km. (32 miles) per hour.

* * *

In order to allow the same speed to be maintained to Bukama, where the Upper Congo navigation commences, the Lower Congo-Katanga Railway Company should relay its line and substitute 40 kgr. (80 lb. per yard) for the 29.1 kgr. (58.2 lb. per yard) rails now in use.

It is true that the existing line carries engines weighing from 85 to 118 t. in working order; the fact remains however, and the experience of the South African Railways appears to be conclusive in this respect, that an average speed of 52 km. (32 miles) per hour requires a stronger track laid with rails of 40 kgr. per metre (80 lb. per yard).

As regards the Upper Congo to the Great African Lakes Railway Company, the incorporation of its Bukama to Stanleyville line in the Trans-African railway can only be realised at the cost of a complete reconstruction.

It will be necessary to abandon river navigation between Bukama and Kongolo and Kindu and Ponthierville, and to construct railways over these sections, connecting with the existing lines from Kongolo to Kindu and from Ponthierville to Stanleyville, thus connecting this last mentioned place directly to Bukama; and further, to take up the whole of the existing track, laid with 24.4 kgr. (49 lb. per yard) rails, and to extend the stronger track with 40 kgr. (80 lb. per yard) rails through to Stanleyville.

When this has been done, the five Southern sections of the Trans-African line, with standard equipment, will form the great colonial trunk line giving through communication without reloading from Cape Town to Stanleyville.

* * *

This work of standardisation, which moreover is necessary for the rapid development of the territory served, is regularly provided for, at any rate as far as the frontier of the Belgian Congo, in the annual estimates of expenditure of the South African Companies. An examination of these estimates is very instructive, and shows the excellent management and flourishing condition of these important Companies.

The « South African Railways » and the « Beira Mashonaland and Rhodesia Railways » publish the results of each working year's in a very original form, the most recent being given hereafter (See figs. 2 and 3).

* * *

The essential fact, shown by the following charts, is that on the two systems the net receipts show an appreciable balance after paying the interest and sinking fund on the first cost of construction. The South African railways are not only self supporting and in need of no financial assistance, but pay an excellent return on the money invested.

As regards the gross receipts: 26.61 % are from passengers and parcels, and 70.39 % from freight of all categories, on the South African Railways, which serve the States of the Union of South Africa, already fairly thickly populated. On the other hand, the « Beira and Mashonaland and Rhodesia Railways ». running through territory in which the population is still scanty, show only 11.3 % for passengers and parcels and 84.8 % for general freight traffic.

* * *

It may now be of interest to give a brief summary of the working expenses of the two companies.

South African Railways.

The coefficient of operation, which was 78.9 % in 1920, fell to 77.52 % in 1924.

For this last year the working expenses are divided as follows:

Way and works	16.18 %
Motive power and rolling stock	46.22 %
Traffic Department	22.50 %
General expenses	6.20 %
Renewal fund for permanent way and rolling stock	8.9 %
Total	100.00 %

Of the 46.22 % for motive power and rolling stock, 23.89 % is for motive power, strictly speaking, 11.91 % for repairs to locomotives and 10.42 % for repairs to rolling stock.

The cost per train-kilometre in 1924 was 4 sh. 9 d.

Finally, in accordance with the graph showing working expenses, 48.52 % are due to the cost of labour. It is true that this figure is the total for the permanent way, motive power and traffic departments, but it is none the less significant and indicates the considerable influence which any increase in wages may have on working expenses.

Beira and Mashonaland and Rhodesia Railways.

In 1920, the coefficient of operation was 68.7 % and has fallen to 56.37 % in 1924.

The working expenses for the last year are divided as follows:

Way and works	17.7 %
Motive power and rolling stock	38.3 %
Traffic Department	18.3 %
General expenses	8.2 %
Renewal fund for permanent way and rolling stock	17.5 %
Total	100.0 %

Under the heading motive power and rolling stock, maintenance of locomotives accounted for 9.3 %, maintenance of rolling stock 6.2 % and motive power strictly speaking, 22.8 %.

Consumption of coal was 18.75 kgr. per engine-kilometre (66.52 lb. per engine-mile); and the cost per train-kilometre was 6 sh. 8 d.

* * *

The South African railways owe their prosperity mainly to their very efficient organisation.

As regards rates, they are guided by the principle laid down in 1917 by the General Manager of the South African Railways:

« In South Africa, where distances are « great and populations spare, the appli- « cation of business principles is essen- « tial, if industries are to be encouraged « and stimulated.

« Local markets are inadequate for « many of our products, and if we are « to find outside markets, it will be at « ruling market rates. »

Experience has shown the soundness of this point of view, and the General Manager of the Beira and Mashonaland and Rhodesia Railways was able to state in his report for 1924:

« It is clear that in a new colony, si- « tuated far from the seaboard, and « struggling to compete in the world's « market for the sale of the products, on « which its future prosperity depends, « the question of cost of transportation « must play an important part.

« The community will be greater bene- « fited and trade will be better stimu- « lated, by applying more substantial « reductions to selected traffics, which « stand most in need of assistance. This « is the principle on which the Railway « Administration has based its policy of « reductions. »

The practical application of such a sound policy has not failed to produce excellent results. The export traffic from the ports of the Union of South

MAP OF THE SOUTHERN SECTIONS OF THE TRANS-AFRICAN RAILWAY.

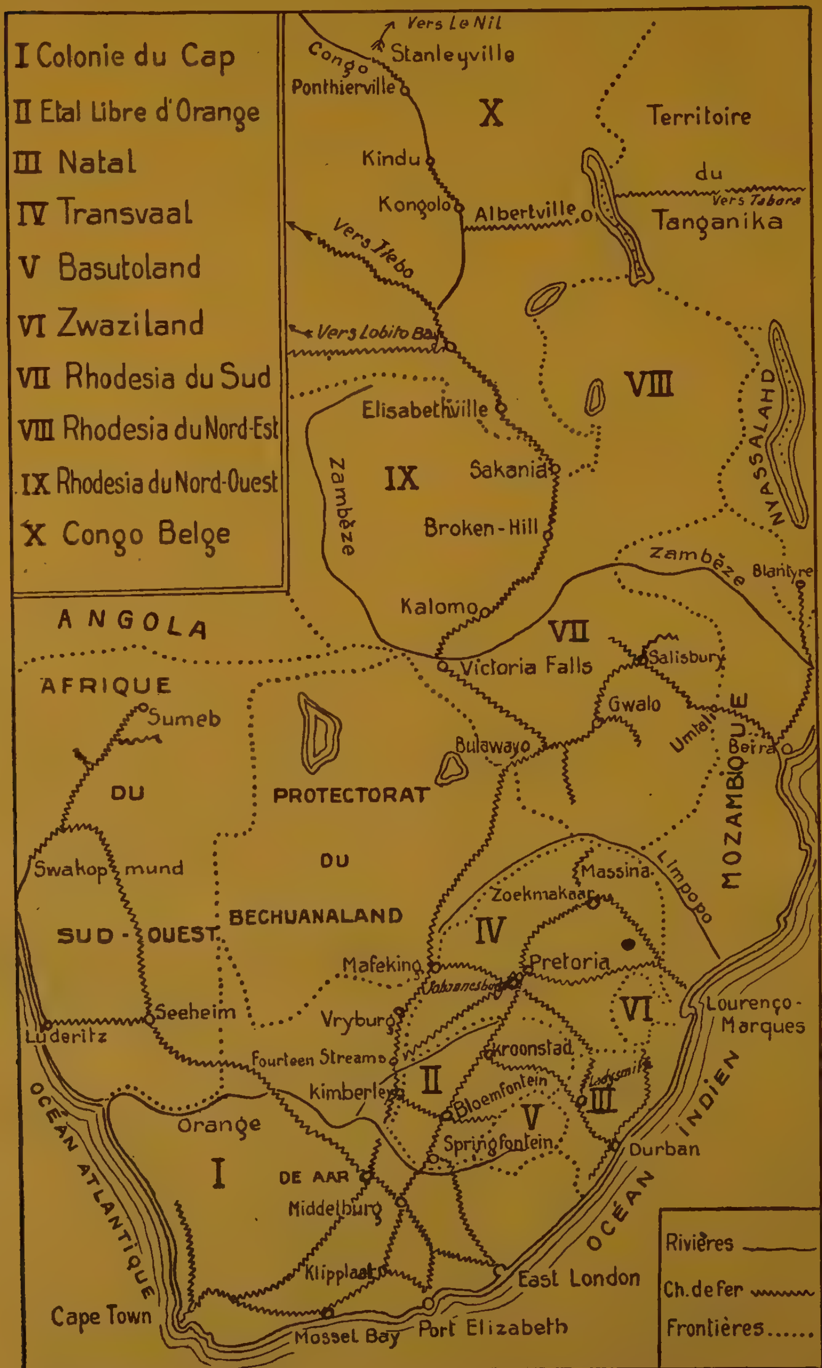


Fig. 1.

Explanatory note : I. Colonie du Cap = Cape Colony. — II. État Libre d'Orange = Orange Free State. — III. Natal = Natal. — IV. Transvaal = Transvaal. — V. Basutoland = Basutoland. — VI. Zwaziland = Swaziland. — VII. Rhodesia du Sud = South Rhodesia. — VIII. Rhodesia du Nord-Est = North-East Rhodesia. — IX. Rhodesia du Nord-Ouest = North-West Rhodesia. — X. Congo Belge = Belgian Congo.

Explanation of French terms : Rivières = Rivers. — Ch. de fer = Railways. — Frontières = Frontiers.

SOUTH AFRICAN RAILWAYS. (Year ending March 1925.)

1. GROSS RECEIPTS.

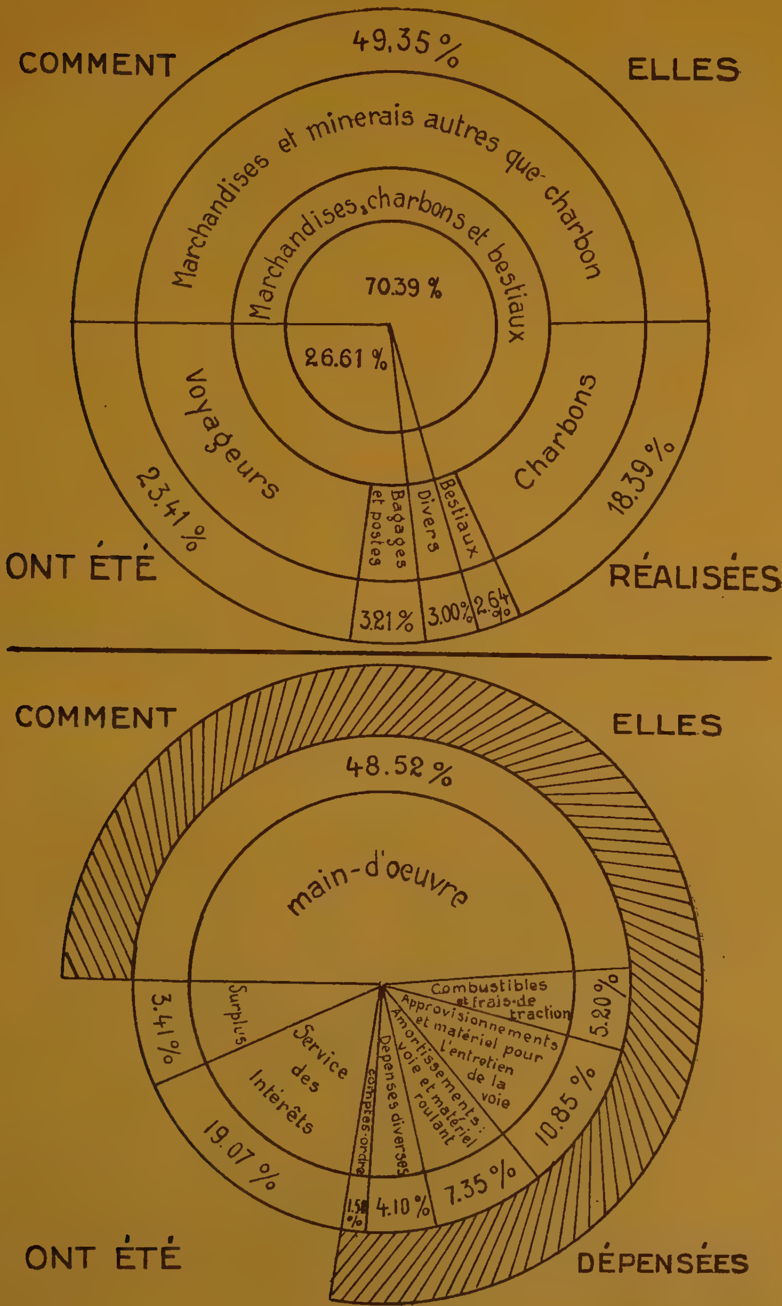


Fig. 2

Explanation of French terms : Comment elles ont été réalisées = How they are made up. — Marchandises et minerais autres que charbon = Freight and minerals other than coal. — Marchandises, charbons et bestiaux = Freight, coal and cattle. — Voyageurs = Passengers. — Bagages et postes. — Parcels and post. — Bestiaux = Cattle. — Charbons = Coal. — Divers = Miscellaneous. — Comment elles ont été dépensées = How expenditure is divided. — Main-d'œuvre = Labour. — Service des intérêts = Interest on capital. — Dépenses diverses = Various expenses. — Amortissements : voie et matériel roulant = Depreciation of permanent way and rolling stock. — Approvisionnements et matériel pour l'entretien de la voie = Material and stores for upkeep of track. — Combustibles et frais de traction = Fuel and motive power expenses. — Comptes ordre = Accountancy.

BEIRA AND MASHONALAND AND RHODESIA RAILWAYS.
(1924-1925.)

GROSS RECEIPTS.

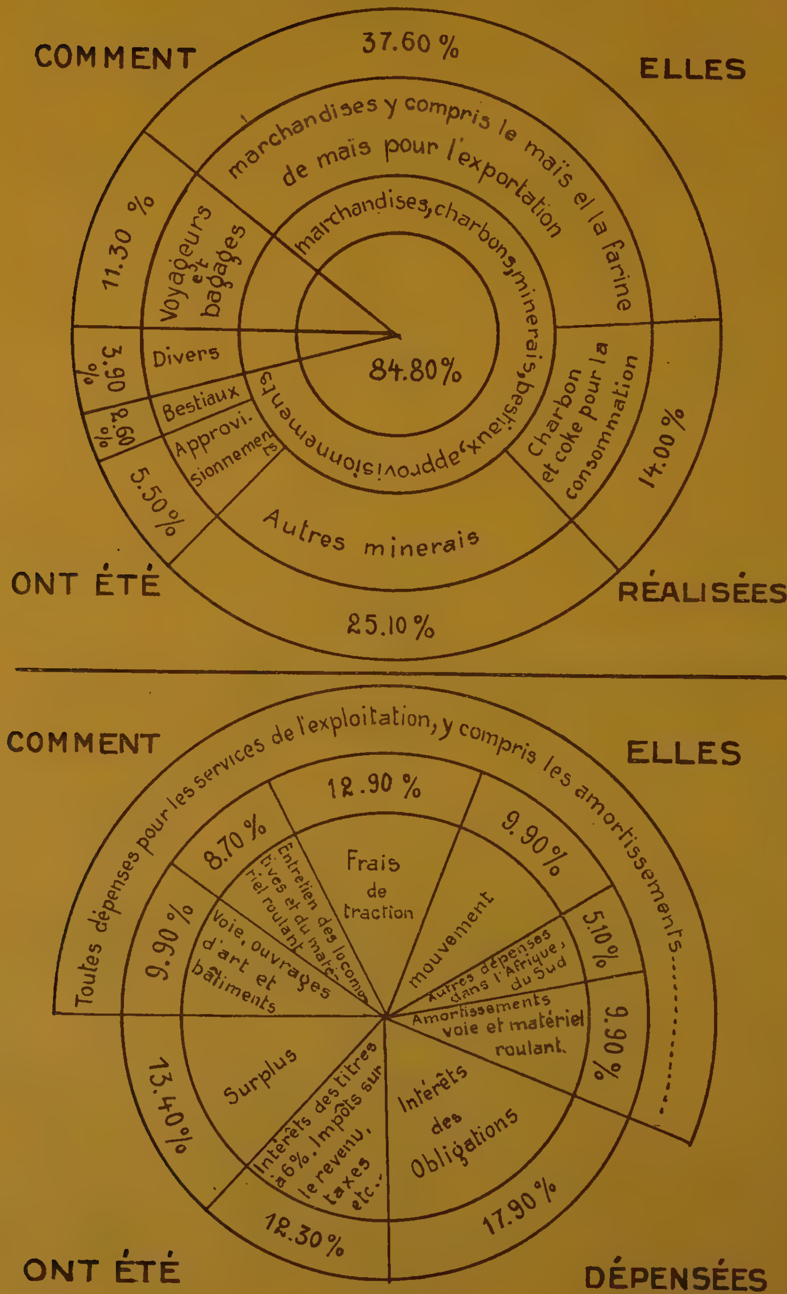


Fig. 3.

Explanation of French terms in figure 3.

Comment elles ont été réalisées = How they are made up. — Marchandises, y compris le maïs, etc... = Freight including maize and maize flour for export. — Marchandises, charbons, minerais, etc... = Freight, coal, minerals cattle, stores. — Charbon et coke pour la consommation = Locomotive coal and coke. — Approvisionnements = Stores. — Divers = Miscellaneous. — Comment elles ont été dépensées = How expenditure is divided. — Toutes dépenses pour les services de l'exploitation, etc... = Total cost of operation, including depreciation. — Voie, ouvrages d'art et bâtiments = Way & Works Department. — Entretien des locomotives et du matériel roulant = Upkeep of locomotives and rolling stock. — Frais de traction = Motive power expenses. — Mouvement = Trafic Department, expenses. — Autres dépenses dans l'Afrique du Sud = Other expenses in South Africa. — Amortissements, voie et matériel roulant = Depreciation permanent way and rolling stock. — Intérêts des obligations = Interest on debenture stock — Intérêts des titres à 6 %, impôts sur le revenu, taxes, etc. = Interest on 6 %, stock, taxation, etc. — Autres minerais = Other minerals. — Bestiaux = Cattle. — Voyageurs et bagages = Passengers and parcels.

Africa, especially from Durban and Delagoa Bay amounted, in 1924, to about 366 000 t., chiefly coal and agricultural produce, which had been brought by rail from distances up to 1 700 km. (1 055 miles). During the same year the port of Beira exported coal from the Wankie Basin at a distance of 1 500 km. (930 miles), and minerals, mainly copper ore, from Katanga, at a distance of 2 500 km. (1 550 miles) to a total of more than 380 000 t.

It may be of interest to mention that on the South African Railways a minimum rate in fares for grain is less than one-tenth of a penny per ton-kilometre. On no other system in the whole world is there yet such a low rate.

Generally speaking, it is these important reductions in rates, for long hauls, and in accordance with special scales or fixed rates for complete wagon loads, which have led to the progressive development of transport, and of paying transport, over the whole of the railways of South Africa.

* * *

What are the conclusions to be drawn as regards the Northern sections of the Trans-African lines yet to be built, from the experience of the Southern sections which have been in full operation for many years?

It will be seen that for distances greater than 1 000 or 1 500 km. (620 or 930 miles), with an annual traffic of at least from 3 to 400 000 t., for traffic over the whole distance, extremely low rates may be fixed without either impairing the financial stability of the system or even preventing a profit from being made after paying all expenses.

This fact has a particular importance as regards a line through desert regions, such as those found over the greater part of the Northern sections of the Trans-African line, that is to say the Trans-Sahara.

This line will connect Algeria to districts of the Sudan such as Nigeria or Chad, which are capable of great agricultural development. Industries quickly develope when produce can be dispatched rapidly and economically by rail. The case would be similar to that of Rhodesia, where it is necessarry to transport many hundreds of thousands of tons of goods a total distance of from 2 500 to 3 000 km. (1 550 to 1 850 miles).

This being so, it is not too optimistic to assume, for the Trans-Sahara line, a coefficient of operation less than 60 % (1), this being based on rates for low value material of 0.02 to 0.03 franc per ton-kilometre, the more valuable freight, such as cotton, being charged at

(1) A direct estimate of the cost of operation places the coefficient at 56.8 %. Increases in passenger fares and in the rates for valuable freight, which are quite admissible in the first place as a provisional measure, should allow this coefficient to be reduced still further.

0.05 or 0.06 franc, or even provisionaly at 0.08 franc.

On the other hand, the net receipts, which may be taken as more than 40 % of the gross receipts, will no doubt be sufficient to carry, if not all, at any rate a very considerable proportion of the capital charges for the construction of the Trans-Sahara railway.

The actual capital outlay will be relatively small by reason of the very few works necessary when crossing the desert. The earth works will be almost insignificant, and there will be neither bridges nor tunnels. Sand can be effectively prevented from being carried on to the line by the wind by simple and inexpensive means; the problem of feed water is now solved, and will not cause any serious difficulty. Finally, the organisation for laying the track, if carefully thought out in all details, will ensure a rapid advance of the rail head, and the work should be finished within seven years. In brief, the cost of construction will not be any greater for the Northern sections of the Trans-African line than for the Southern sections, the influence of local conditions having been taken into account.

* * *

In any case, by considering the example offered by South Africa, and especially by Rhodesia, in the matter of railway transport, a very clear impression will be obtained. One can see what can be done by enterprise rationally applied and how bold initiative leads to success.

France should loose no time in recognising this, and should no longer hesitate to carry out a work upon which depends the more intensive development of her vast domains across the Mediterranean, as, up to the present, she has done for fear of incurring too great financial liabilities.

The Trans-Sahara railway, or rather the Northern sections of the Trans-African railway, even if they may perhaps necessitate temporary sacrifices, would not impose any permanent cost on the State; they would undoubtedly soon become a source or revenue owing to the participation of the State in the net revenues.

APPENDICES.

(1) The South African Railways and Harbours administrate the whole of the railways of the Union of South Africa, including the former German South-West Africa. The system is made up as follows :

Cape Colony	6 986 km.	(4 340 miles).
Natal	2 299 —	(1 428 —).
Transvaal	4 295 —	(2 669 —).
Orange Free State	2 159 —	(1 342 –).
Total.	15 739 km.	(9 779 miles).
Former German South-West Africa	2 142 km.	(1 331 miles).
Private lines	910 —	(565 —).
Total system administrated by the South African Railways and Harbours	18 791 km.	(11 675 miles).

Excluding the first Southern section of the Trans-African line, from the Cape to Vryburg, this system includes the following main lines, from which radiate branch lines :

Central Line : From the Cape via De Aar, Kimberley, Bloemfontein and Johannesburg to Pretoria.

Branch lines to the ports :

Eastern ports of the Union of South Africa: Pretoria to Lorenzo-Marquez;

Johannesburg and Kroonstad via Ladysmith to Durban;
Bloemfontein to East London;
De Aar to Port Elizabeth;
De Aar to Mosselbay.

Western ports of former German South-West Africa :

Luderitz, via Seeheim and Nakop to De Aar;

Swakopmund (Walvis Bay) via Seeheim and Nakop to De Aar.

(2) The Beira and Mashonaland and Rhodesia Railways operate the whole of the Rhodesian railways and the junction line to the port of Beira through Portuguese Mozambique.

The total length of line being 3 961 km. (2 461 miles), made up as follows :

Trans-African :		
2nd Southern section :		
Vryburg to Bulawayo	946 km.	(588 miles).
3rd Southern section :		
Bulawayo to Sakania	1 266 —	(787 —).
The Rhodesia Railway Company, Limited :		
Bulawayo to Salisbury	481 —	(298 —).
Branch lines including the Blinkwater Railway Company, Limited	415 —	(258 —).
The Mashonaland Railway Company, Limited :		
Salisbury to Umtali (frontier of Portuguese Mozambique)	274 —	(170 —).
Branches	251 —	(156 —).
The Beira Railway Company, Limited and The Beira Junction Railway Company, Limited :		
Umtali to Beira (Portuguese Mozambique)	328 —	(204 —).
Total length of the Rhodesian Railway system.	3 961 km.	(2 461 miles).

(3) The Lower-Congo-Katanga Railway Company's system, excluding the Katanga line itself — which forms the 4th Southern section of the Trans-African railway — has two main extensions :

1st. The line from Lower-Congo to Katanga, under construction between Bukama and Ilebo on the Kasai — 1 100 km. (685 miles) — and under survey between Ilebo and Kinshasa;

2nd. The proposed junction line from the Katanga to the Benguella Railway.

This line will leave the Katanga railway at Fungurume, between Kambove and Bukama, and be carried to the Eastern frontier of Angola near Dilolo, to connect with the Portuguese line from Lobito Bay.

(4) The present system of the « Upper Congo to the Great African Lakes Railway Company », — the 5th and last Southern section of the Trans-African Railway, — is completed by the line which runs from Kabalo, on the Upper Congo, 565 km. (351 miles) down stream from Bukama and 75 km. (46.6 miles) up stream from Kongolo, to the port of Albertville on Lake Tanganyka, a distance of 273 km. (170 miles).

The Company also operates the lake steam ship service : the direct crossing in an Easterly direction to Kigoma of 130 km. (81 miles); touching at the various ports when northbound to Uvira, 340 km. (211 miles) away, and when southbound to Kituta 465 km. (289 miles) away.

The Company is also carrying out a final survey for the location of a railway from the Congo to the Nile, between Stanleyville, and Mahagi on Lake Albert Nyanza.

[628 .142.2 & 691]

Percentage renewals and average life of railway ties,

By J. D. MacLEAN.

ENGINEER IN FOREST PRODUCTS, U. S. FOREST PRODUCTS LABORATORY, MADISON, WIS.

Figs. 1 to 4, pp. 111 to 114.

(From *Engineering News-Record.*)

For a number of years the Forest Products Laboratory, in co-operation with the railroads, has been collecting tie-service records from test sections of track in various parts of the country. These records comprise both treated and untreated ties of different species. Inspections are made at regular intervals (usually annually) after the ties are placed in service and renewals started. When all the ties in a test section have been removed, the average life of the ties in the group is computed.

Obviously the average life will vary widely for different groups. Nevertheless, throughout all of the groups there is a remarkably constant correlation between the proportion of renewals made and the corresponding elapsed proportion of the average life. To illustrate, in a group of ties having an average life of 12 years, about 22 % would be renewed in 9 years, and in another group having an average life of 8 years about 22 % would be renewed in 6 years, or at 75 % of the average life in each case.

Renewal curves. — Several years ago the curve *A* in figure 1 was prepared by M. E. Thorne, showing the percentage of tie renewals plotted against percentage of average life elapsed (see *Proceedings of the American Wood Preservers' Association,* 1918). This graph takes the form of the « cumulative frequency » or « summation » curve derived from the familiar probability type, and is based on completed service test records of 12 185 untreated ties from 26 groups, and 30 751 treated ties from 17 groups, or 42 936 ties in all. A number of railroads in using this curve have found that it indicates very closely the relation between renewals and average life of ties in their own practice, especially when the number of ties involved is fairly large.

Since the preparation of this curve, a large number of other service tests have been completed, and curves based upon these indicate clearly the operation of the law of probability in the replacement required annually. The additional records include 44 970 untreated and 39 646 treated ties, making a total of 57 155 and 70 397 respectively, or a grand total of 127 552 ties. There were 58 different groups of untreated and 37 groups of treated ties, or 95 records altogether. The records, which include renewals for all causes, are probably the most extensive thus far obtained from the standpoint of numbers, distribution, and reliability. An effort was made to eliminate all data not considered entirely reliable.

For each group of ties the average

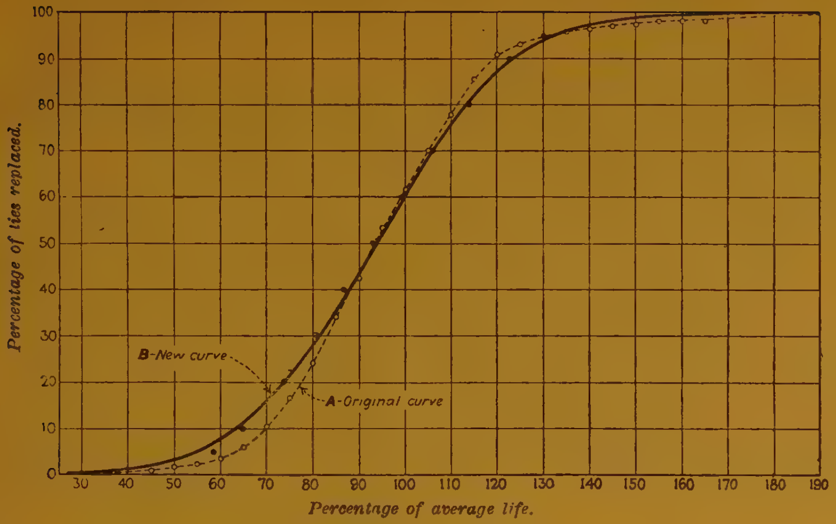


Fig. 1. — Comparison of tie renewal curves.

A = Original curve based on records of 42 936 ties. — *B* = New curve based on records of 127 552 ties.

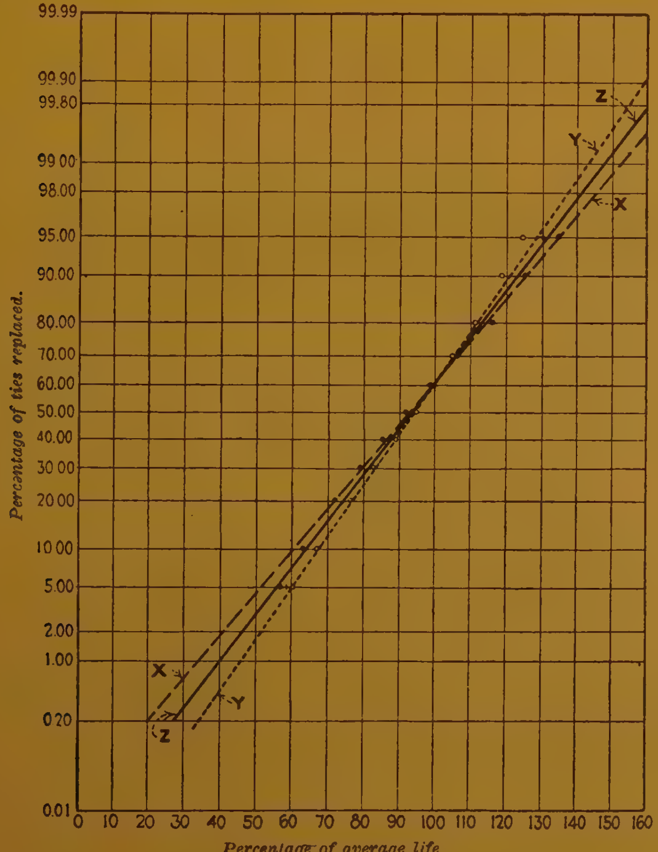


Fig. 2. — Relation of tie replacement to average life.

Curve X for 70 397 treated ties. — Curve Y for 57 155 untreated ties. — Curve Z for all the 127 552 ties.

life, the percentage of renewals, and the percentage of average life elapsed at each renewal period were computed. Curves were drawn for the individual groups, showing the relation between the total percentage of tie renewals and the percentage of average life elapsed. Readings were then taken from each of the curves (for definite percentages of renewals) and a weighted average value of percentage of average life elapsed was computed. Weighted averages were determined for untreated ties alone, treated ties alone, and for all ties including both treated and untreated. The data thus obtained are shown in the accompanying table.

Summation curves. — The information given in this table is plotted in figures 1 and 2. Figure 1 shows both the original curve, *A*, mentioned above and the corresponding new curve, *B*, derived from the more complete data on 127 552 ties. It will be observed that the results indicated by the former are fairly well checked, over the more important range, by the latter curve.

Figure 2 shows the summation data of the table plotted on « probability » paper. This paper is so constructed that when data which vary in accordance with the normal law of probability are plotted in the summation form as functions of their probability of occurrence, the points will lie on a straight line. With the data thus plotted it is possible to extend the curve and study the results that may be expected beyond the range over which data are obtained. It is also more convenient to use the straight line than the curve plotted on rectangular co-ordinate paper, as in figure 1, especially at the upper and lower limits.

Separate and combined curves. — There is apparently some difference in the relation of renewals to average life for the treated and untreated ties, shown in figure 2, more particularly in the range where the renewals are small. In the more important working range, however, the difference is only slight. It is possible that the variation arises on account of decay being the more important influence affecting renewals of untreated ties, whereas mechanical wear is the more potent factor with treated ties. In any event the small difference in the two curves makes it possible to employ a combined curve with sufficient accuracy.

In the combined curve for all 127 552 ties the points fall practically along a straight line through the co-ordinate point representing 50 % renewals and 94 % of average life. The results are strikingly close to a true probability relation, as proved by the fact that a straight line can be drawn through a majority of the plotted points.

By extending the line of the combined curve it will be found to intersect the horizontal top line (99.99 % replacements) at about 181 % of the average life. In other words, practically all ties are replaced when about 181 % of the average life has elapsed. Computations made from the data to determine the percentage of average life at which all ties were removed gave about 180 %, which checks the curve very closely. An examination of the lower end of the combined curve shows that nearly 40 % of the computed average life elapses before 1 % of the ties are replaced. Renewals then become more frequent, reaching a maximum rate between 90 and 100 % of the average life.

Use of curve diagram. — The following example will illustrate the use of the single straight-line combined graph. Assume that in a homogeneous group of ties, all placed in service at the same time, 25 % have been removed in 6 years. Using the graph, it will be noted that 25 % would normally be removed when

about 78 % of the average life has elapsed. Hence six years is 78 % of the average life to be expected, which is thus figured at about 7.7 years.

If it is assumed that the group contained 1 500 ties and it is desired to estimate the removals expected during the succeeding year, take 7 years as a percentage of 7.7 years, the average life. The result is 91 %, and the curve shows that a total of 45 % or 675 ties will have been replaced at that period. Renewals to be expected during the seventh year, therefore, amount to 300 ties. The renewals for subsequent years can be readily computed in the same manner.

If in the above case the curve for untreated ties had been used, the computed average life would have been about 7.5 years. The curve for treated ties would have given about 7.8 years. In either case the variations from the figures obtained by using the combined curve would be less than 3 %. Since the graph is based on extensive data from test groups of ties placed in many different sections of the country and under widely varying traffic conditions, it is felt that it furnishes a reliable guide for estimating the average life and the rate of tie renewals when the group of ties under consideration is large enough to be representative.

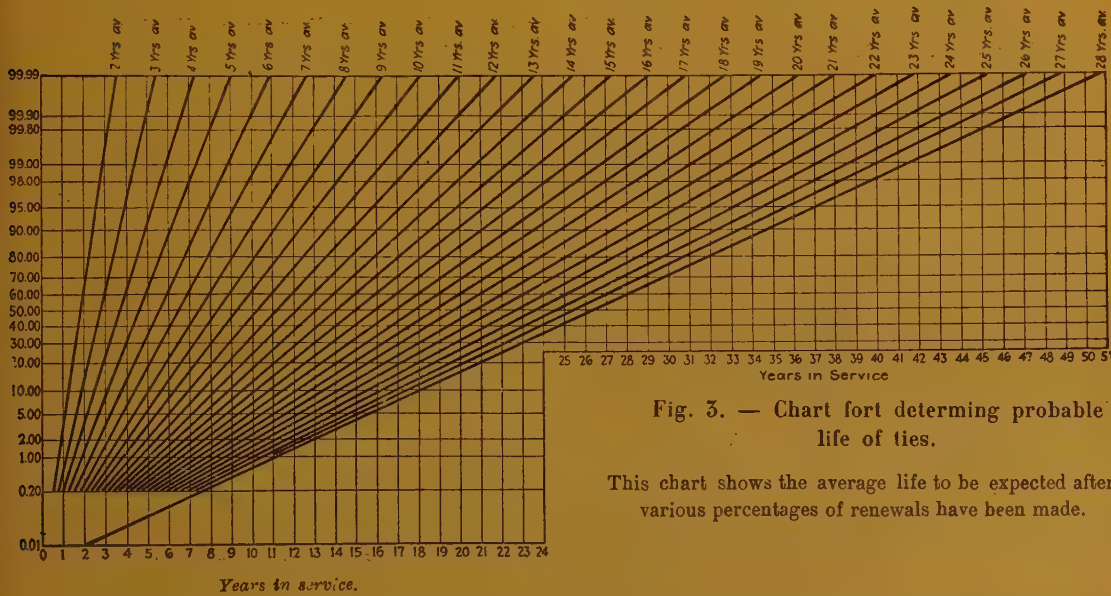


Fig. 3. — Chart fort determing probable life of ties.

This chart shows the average life to be expected after various percentages of renewals have been made.

Chart for simplified computation. — In order to avoid the computations necessary in using the curves shown in figures 1 and 2, the chart figure 3 has been prepared, which is based on the curve for all ties in figure 2. Probability paper was used for this chart also. However, in this case definite average life values were assumed and the percentage of ties replaced (corresponding to different proportions of the average life) was plotted against years in service instead of percentage of average life, as in figure 2. The diagonal lines are marked with the average life to be expected for various percentages of repla-

cements after a given period of years in service.

Method of using chart. — The previous example will illustrate the use of this chart. For a group of ties, 25 % are assumed to be removed after 6 years. The horizontal line representing 25 % replacements intersects the vertical 6-year line at about 7/10 of the horizontal distance between two diagonals, one representing 7 and the other 8 years average life. The point of intersection, therefore, represents an average life of 7.7 years, checking the value previously determined from figure 2.

To determine subsequent renewals, let it be assumed again that there are 1 500 ties in the group and that 25 % are removed after 6 years service. The total percentage of renewals to be expected by the end of the 7th year can be found by reading upwards on the 7-year line to a point about 0.7 of the horizontal distance between the diagonals representing an average life of 7 and 8 years, respectively. This is about 45 %, or a total of 675 ties, checking the result found previously from figure 2. Then 675 less 375 gives 300, the number to be replaced during the 7th year. Likewise, reading up the line of 8 years service to a point 0.7 of the horizontal distance between the 7 and 8-year average life diagonals, it is found that about 65 %, or 975 ties, are removed by that time. Similarly, it will be found that a total of about 84 % or 1 260 are removed in 9 years. The time at which practically all ties are out can be determined by reading on the horizontal scale directly below the point on the top line taken 0.7 the distance between the 7- and 8-year diagonals. Projecting from this point to the lower scale, it will be found that practically all ties of this group, having a computed average life of 7.7 years, would be removed in slightly over 13.9 years, or approximately 14 years.

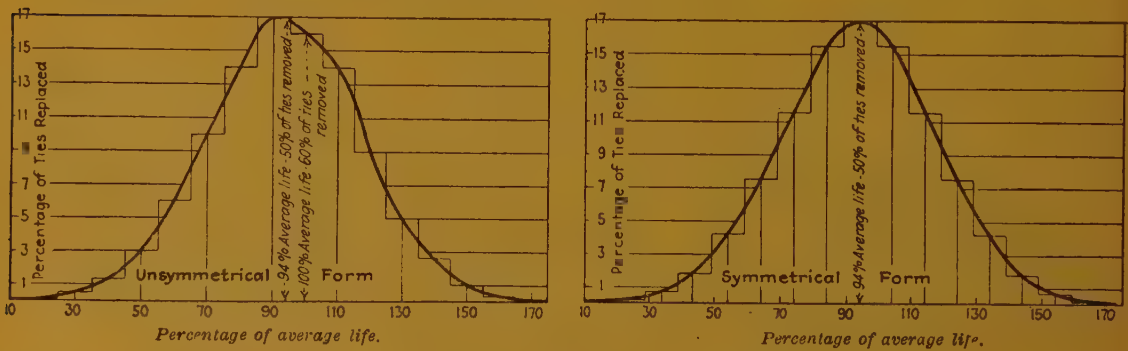


Fig. 4. — Frequency curves for tie replacements.

These curves show successive percentages of the replacements for 10 % intervals of average life. The curve at the left is of unsymmetrical form, with origin taken at 100 % average life. The curve at the right is of symmetrical form, with origin taken at 94 % average life.

Frequency curve. — If a frequency curve (showing percentages of ties removed at successive time intervals) is plotted from the combined data with the origin of co-ordinates of the curve at 100 % average life, as at the left in figure 4, a certain degree of positive skewness will be found. This is be-

cause more than 50 % of the ties have been replaced before 100 % of the average life is reached, as already indicated by the cumulative curve in figure 2.

Data for cumulative frequency curves.

Untreated 57 155 ties.		Treated 70 397 ties.		Treated and untreated combined 127 552 ties.		Percentage average life from line Z-Z (fig. 2).
Percentage of renewals.	Percentage of average life.	Percentage of renewals.	Percentage of average life.	Percentage of renewals.	Percentgae of average life.	
5	60.9	5	56.7	5	58.5	55.0
10	67.3	10	63.1	10	64.8	63.8
20	76.3	20	72.1	20	73.9	73.6
30	82 7	30	79.3	30	80.7	81.5
40	88.8	40	85.3	40	86.8	88.0
50	94.5	50	91.9	50	93.1	94.0
60	99.9	60	98.8	60	99.2	100.0
70	105.6	70	106.5	70	106.0	106.0
80	111.7	80	116.0	80	114.1	113.5
90	119.1	90	125.7	90	122.7	123.5
95	124.5	95	134.5	95	130.0	132.0

It is very likely that the following are factors of track maintenance which tend to produce skewness of the curve with respect to the origin taken at 100 % of the computed average life :

A. — In the early removals, consideration of economy may make it desirable to remove a certain number of ties that would normally be serviceable for a longer period.

B. — When a large proportion of the ties have been replaced, those surviving have no noteworthy influence on the track conditions. For that reason some of them may be left in service a considerably longer time than would be permissible if ties of the same condition were present in large numbers. Even a few ties left in service several years longer than the majority of those in the group may have considerable weight toward increasing the computed average life. This would tend, of course, to produce an unsymmetrical frequency curve when the origin is taken at 100 % average life.

C. — Other factors which doubtless increase the early mortality of ties are fires, washouts, wrecks, etc.

To eliminate skewness in the graph, the origin may be taken at the « mode » (94 % average life) instead of the arithmetic mean (100 % average life). By this means a normal frequency curve applying correctly to the data will be obtained. Such a curve is given at the right in figure 4, which shows the rate of replacements for 10 % intervals of average life. The renewals for successive 10 % intervals are represented by rectangles, and a smooth curve is drawn through the midpoints. It will be noted that between 29 and 39 % of the average life only 0.70 % of the ties are removed, whereas between 89 and 99 % of the average life (the period of maximum replacements) 17 % are removed.

In the equation for the symmetrical

form of the frequency curve in figure 4 as given below Y represents the frequency or percentage of replacements for a given value x; x is the distance to the right or left of the origin measured in intervals of 10 % average life for the group considered; e is the base of the Naperian system of logarithms.

$$Y = 17e^{-\frac{x^2}{10.95}}$$

General considerations. — In the use of all the foregoing curves the following considerations should be borne in mind:

1. — The larger and more homogeneous the group of ties under consideration, the greater the likelihood that the curve of renewals will closely represent the results that will be obtained.

2. — When the number of renewals is very small, the average life estimated from the curves will not be as reliable as that indicated when larger proportions of renewals have been made. When, however, the number involved is sufficient to furnish a representative sample, the chart or curves provide a means of closely estimating the rate of renewals and average life to be expected, which will hold for groups of ties having either a long or a short average life.

[621 .135. (01 & 621 .135.4]

Note on the stability of various types of locomotives,

By Professor G. QUAGLIA, Engineer.

Figs. 1 to 14, pp. 120 to 136.

(*Rivista dei Trasporti.*)

1. — The tendency to increase the speed of main line trains necessitates the use of transition curves to remove the risk of derailment and to protect passengers from unpleasant shocks.

It is wellknown, in practice, that when transition curves are not used, the vehicles are subjected, at the commencement and end of a circular curve, to a rolling oscillation due to the sudden action of the centrifugal force and to a lateral oscillation caused by the change in the direction of motion. These oscillations cause the flanges of the wheels to exert considerable pressure against the outside rail. In order to decrease this it is usual to balance the centrifugal force, either totally or partially, by superelevating the outside rail above the inside rail throughout the whole length of the circular curve, and in order that this superelevation may increase gradually, from zero on the straight line up to its final value, it is usual to interpose between the straight line and the circular curve a transition curve.

For this purpose the type of curve used may be the clotoide, the lemniscate or the cubic parabola; the cubic parabola is the most generally used in view of the ease with which it may be set out.

Until latterly, the somewhat high figures of from 3 to 3.5 % have been adopted for the inclination of the outside rail over the inside rail, and this has resulted in rather short transition curves. In our « Instructions on the laying of the track », which date back to 1908, curves of a length of 40 m. (43 7 yards) even for speeds of

100 km. (62 miles) per hour are permitted.

However, Professor R. Petersen [1] considers that for high speeds the length of the transition curves now in use should be increased so that the trains may pass easily round the curves, and, from the results of trials carried out on overhead railways, he expresses the opinion that transition curves should not be passed over in less than 3.6 seconds, and that consequently the length of these curves may be given by the relation :

$$l \geqslant V \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where l is expressed in metres and V is the speed in kilometres per hour.

On the other hand, Mr. Descubes, the Permanent Way Engineer of the French Eastern Railway [1], considers that the inclination i of the outside rail above the inside rail on transition curves should be decreased as the speed increases, and that consequently the length of the transition curves should increase with the speed.

He recommends :

For V = 60 km. (37.3 miles) per hour and R = 300 m. (328.1 yards), $i = 0.0016$,
For V = 120 — (74.6 miles) — and R = 800 m. (874.9 yards), $i = 0.001$.

There is, however, evidence, which is very important from a practical point of view, that these authors have not considered the problem from every aspect; in fact they have only taken into account the track, whereas the oscillations to which rolling stock is subjected on curves depends not only upon the track, but also on the characteristics of the rolling stock itself.

The credit for having investigated the stability of rolling stock is due to the French engineer Georges Marié, the author of the wellknown articles on the oscillations of rolling stock and of a recent and very interesting treatise on the same question [2].

In order to call attention to the effects the construction of the track and of the rolling stock have on the stresses borne by the track and on good riding, we will give a brief resumé of Mr. Marié's investigations as applied to a comparison of three locomotives of the *Pacific* type the fastest engines in service on the Italian railways, which differ as regards their suspension and lateral control springs. The results at which we shall arrive, in conjunction with those obtained experimentally, will allow us to come to certain conclusions affecting the problem of the increase of train speeds.

As for curves of more than 1 000 m. (50 chains) radius, it is agreed that the superelevation of the outside rail should decrease progressively in the straight track which precedes the tangent point, we will consider in turn the stability of rolling stock on straight track with the outer rail superelevated, and on curves.

II. *Stability on straight track.* — To calculate the coefficient of security against derailment of a pair of wheels, we may remark that when one of the wheels makes a positive angle of friction with the rail, a condition which tends to cause derailment, the limiting value of the force capable of causing derailment is given by :

$$F < \Pi \frac{\tan \beta - \varphi}{1 + \rho \tan \beta} \quad . \quad . \quad . \quad (2)$$

where Π is the weight carried by the

[1] R. Petersen : *Die Gestaltung der Bogen im Eisenbahngleise* (The form of railway curves). Kreidel, Berlin and Wiesbaden, 1920.

[2] G. Marié : *Traité de stabilité du matériel des chemins de fer* (Treatise on the stability of railway rolling stock). Paris et Liége, and 1924.

[1] M. Descubes : « Note relative aux raccords des courbes et des alignements » (Note on transition curves) (*Revue Générale des Chemins de fer et des Tramways*, June 1922).

wheel under consideration, β is the angle which the common tangent of the surface of the flange and the rail at the point of contact makes with the horizontal, φ is the coefficient of friction between the flange and the rail.

However, the force F on a straight track is as a rule produced by the nosing action which is essentially due to the play existing between the flanges and the rails.

The maximum disturbing force which gives rise to this oscillation is expressed by $P\varphi\varepsilon$ where P is the weight of the vehicle, φ the coefficient of friction between the wheels and the rails, and ε is the play between the flanges and the rails. This assumes that the frictional resistances which developed during the sinoidal movement of nosing are all driving forces. However, as part may be static friction, it is necessary to introduce a coefficient of reduction K which, for speeds of from 100 to 120 km. (62 to 75 miles) per hour, may be taken as 0.50 for vehicles and locomotives of old types, as 0.40 for long locomotives, and as 0.30 for bogie locomotives and vehicles. Based on experiments, Mr. Marié states that the reaction of the rails against the front portion of the vehicle is double that on the rear portion, and consequently the shock may be divided in the ratio of two-thirds on the front and one-third on the rear. It is therefore the reaction on the front portion which concerns us. As the vehicles when forming a train and with their buffers in contact are only slightly subject to nosing, we will consider the case of a *Pacific* type locomotive with the leading bogie having a considerable amount of resistance to lateral displacement due to the presence of strong centreing springs and an appreciable frictional resistance, so that the bogie by itself may be capable of taking the shock. The laterally rigid portions of the bogie, such as the axles and the frame, give rise to a shock which, for speeds not exceeding 120 km. (75 miles) per hour, is not very violent and which is damped out by the lateral elasticity of the track and of the bogie frame. The remaining effect of the shock, when the period of oscillation due to nosing action and that of the horizontal oscillation due to the lateral control springs, do not synchronise, is absorbed by the lateral displacement of the bogie; if, on the other hand, owing to the elasticity of the bogie side control springs synchronism occurs, the amplitude of the oscillations will increase until the resistance caused by the friction of the lateral control devices is equal to the energy of the disturbing forces.

If d is the lateral displacement of the bogie on either side of its normal position, the total lateral displacement during a simple oscillation while the vehicule moves across from one rail to another is equal to $2d$. Consequently, if Q is the weight carried by the bogie and φ' the coefficient of friction of the cross slide, we have :

$$\frac{2}{3} KP\varphi\varepsilon = 2dQ\varphi' \quad . \quad . \quad . \quad (3)$$

Finding the value of d from this equation and knowing the initial force and the strength of the springs, we at once obtain the value of the lateral reaction F.

This is greater than the true value, because the oscillations are supposed to be damped out in the plane of the bogie lateral central springs. The shock is taken, however, at the level of the rails, which causes the spring borne body to roll and part of the energy of the shock is absorbed by the carrying springs, and should synchronism take place, by friction in these springs.

From what has been already said, it will be seen that a bogie of this type serves not only to facilitate the passage of the locomotive round curves, but also to damp out the lateral oscillations.

If the bogie is of the American swing link type, the leading coupled axle is the one subjected to the shock : in this case the vehicle behaves as if it were rigid in the transverse direction. In his article

entitled : « Oscillation de lacet des véhicules de chemin de fer » (Nosing oscillations of railway vehicles) (*Annales des Mines*, 1st volume 1909), Mr. Marié says that the effects of this shock are similar to the effects of a finite mass which collides with an infinite mass, at a distant point on the line parallel to the direction of movement of the striking mass and passing through its centre of gravity.

The kinetic energy produced by the shock is equal to

$$\frac{1}{2} Mv^2 \left[\frac{1}{1 + \left(\frac{h}{\rho}\right)^2} \right]$$

where M is the mass of the moving body, v its speed in metres per second before impact takes place, h the height of the centre of gravity of the moving mass above the level of the rails, ρ the radius of gyration of this mass about the horizontal axis perpendicular to the direction of impact and passing through its centre of gravity.

For the majority of locomotives, the ratio $\frac{h}{\rho}$ is equal to about 2, and consequently the energy expended during the first phase of the impact is equal to about one-fifth of the energy of disturbance. It amounts generally to a few kilogram-metres and is absorbed by the lateral elasticity of the track and of the locomotive. As a result of the impact, the flange of a tyre which comes into contact with the rail, after recoiling slightly and sliding on the rail, again comes into contact with the rail, thereby consuming another fifth of the energy of disturbance. Under the action of lateral pressure against the rail, which in the second phase of the impact first increases and then gradually decreases, the spring-borne mass of the engine turns about its centre of rolling motion. The springs are deflected on the side corresponding to the rail against which the wheel bears, and when synchronism takes place, the amplitude of the oscillation increases until the energy dissipated by friction between the leaves of the springs is equal to the remaining energy of disturbance.

We therefore have :

$$\frac{2}{3} 0.60\ \mathrm{KP}\varphi\varepsilon = \mathrm{P}'f2z \quad . \quad . \quad (4)$$

where f (1) is the *relative friction* which has to be overcome for the springs to deflect under a load P′, and z the variation of deflection of the springs on the same side of the vehicle.

Mr. Marié obtains the maximum value of the lateral force by considering the rotation of the spring-borne portion of the engine around the centre of oscillation and equating the moment of the reactions about the axis of rolling to the moment of inertia of the spring-borne portion about this axis multiplied by the maximum angular velocity. Having found this, it is possible to calculate the lateral velocity and the corresponding acceleration of the centre of gravity of the spring-borne portion, and hence to obtain the force F; it should be noted that in steam locomotives the non spring-borne mass effected by impact is small when compared with the spring-borne portion. The final result is as follows :

$$\mathrm{F} = \frac{1}{6} \cdot \frac{0.60\ \mathrm{KP}\varphi\varepsilon}{fa} \cdot \frac{m}{n} \quad . \quad . \quad (5)$$

where a represents the static deflection of the spring, m half the transverse distance between the springs and n the height of the centre of gravity of the spring-borne portion above the centre of oscillation.

This equation shows us that F decreases as the height of the centre of gravity of the spring-borne portion above the centre of oscillation increases.

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(1) According to Mr. Marié, $f = 2\ \varphi\ (n - 1) \frac{c}{l}$, where φ is the coefficient of friction between the leaves of the springs, n the number of leaves, c their thickness and l the length of the back plate between centres.

Having calculated the value of F, we have now to determine the pressure Π exerted by the wheel against the rail.

As a result of the deflection of the rails at the rail joints, vertical oscillations of the spring-borne portion are set up with, in consequence, variations in the loads upon the springs.

In the most unfavourable case, when synchronism occurs, that is to say, when the oscillations due to gallop synchronise with the time taken to run one rail length, the relative variation in the load on the springs, where equalising levers are not fitted, is given by:

$$D \leqslant \frac{h}{a} + f \quad . \quad . \quad . \quad . \quad (6)$$

If the oscillations synchronise, we have in all cases :

$$h \leqslant 4fa, \quad . \quad . \quad . \quad . \quad (7)$$

where h is the deflection of the rail joints under the passing load and the coefficient 4 is used in the case of one or two axles and rails from 10 to 15 m. (32 ft. 9 3/4 in. to 49 ft. 2 1/2 in.) long. If the number of axles is greater than two, this coefficient is considerably increased, that is to say, the tendency of the oscillations to synchronise is much greater.

From the foregoing expressions it will be seen that this tendency increases and the relative variation of the loads on the springs decreases as the springs are made more flexible.

If there are N axles, connected by compensating levers, the relative variation of the load on the springs becomes :

$$D < \frac{h}{Na} + f \quad . \quad . \quad . \quad . \quad (8)$$

From this the coefficient of security against derailment on a straight track, with the rails on the same level, can be calculated, but when the outside rail is superelevated above the inside rail, two disturbing influences have to be taken into account : the one statical due to the wheels of the vehicle bearing against an oblique surface, and the other dynamical, caused by the sudden rotation of the vehicle at the beginning and end of the superelevated portion of track.

Let us consider the first cause of disturbance by supposing a symmetrical vehicle with two equally loaded axles at a distance apart equal to b, on a level portion of line ABCD (fig. 1). Let us now suppose

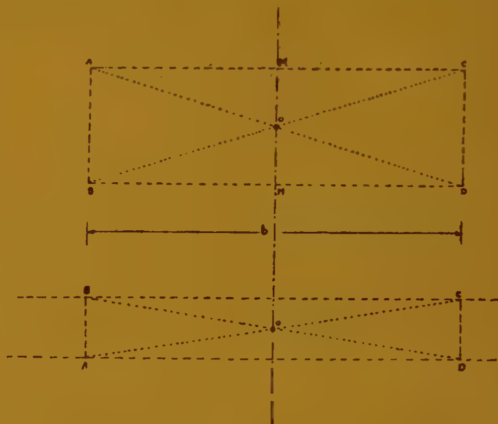

Fig. 1.

that the points M, N, remain fixed, but that the rail AC is inclined $\frac{i}{2}$ upwards and the rail BD inclined downwards. The two rails will therefore have a relative inclination of i.

In this case the buckle of the spring at A is lowered relatively to the horizontal plane passing through MN, by the quantity :

$$AM \cdot \frac{i}{2} = \frac{bi}{4};$$

but if m is half the distance between the springs and s the gauge of the track, the relative decrease in the load in given by :

$$\frac{bi}{4a} \cdot \frac{m}{s}.$$

Let us make the same reduction at D and the same increases at B and C.

The vehicle takes up a position of equilibrium so that the sum of the reactions of the springs about point O is equal to zero.

To find the relative variation Δ of the pressure of the wheels upon the rails, it is sufficient to multiply the foregoing expression by $\frac{m}{s}$, and we therefore have :

$$\Delta = \frac{bi}{4a} \cdot \frac{m^2}{s^2} \quad . \quad . \quad . \quad . \quad . \quad (9)$$

These expressions are also applicable to a vehicle having any even number of axles, provided that it is symmetrical about its mid axis MN and that the static flexibility of all the springs is the same, and as an approximation, these equations may also be used for a locomotive.

The above equation shows that Δ increases as i increases, and decreases if the flexibility of the outer springs increases or if b is decreased, that is to say, if the vehicle is given greater flexibility in a longitudinal direction by means of a suitable spring gear.

Let us now consider the dynamic disturbance.

When entering the section with superelevated outside rail, the vehicle, previously vertical, commences to turn with an angular velocity ω, about a point C (fig. 2), and acquires kinetic energy as given by :

$$\frac{1}{2} I\omega^2$$

where I is the moment of inertia about a longitudinal axis passing through C.

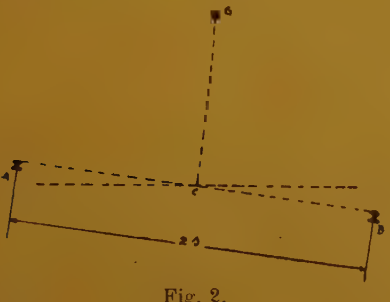

Fig. 2.

If v is the speed of the vehicle, we have :

$$\omega = \frac{vi}{2} \frac{1}{s}.$$

Let P be the weight of the vehicle, ρ its radius of gyration about a horizontal axis parallel to the track passing through the centre of gravity G, and h the height of the centre of gravity above C; we then have :

$$\frac{1}{2} I\omega^2 = \frac{P}{2g}(\rho^2 + h^2)\frac{v^2 i^2}{4s^2} \quad . \quad . \quad . \quad (10)$$

Under the action of this kinetic energy of rotation, the spring-borne portion oscillates about the centre of rolling. The amplitude of the oscillation is such that the strain energy stored in the springs is equal to this kinetic energy.

Taking for the sake of simplicity the case where the non-spring borne portion is a very small part of the total spring-borne weight, and the internal friction of the springs is negligible, which assumption may safely be made, as there is no danger of synchronism occurring in this case, the springs on each side of the vehicle will carry the load $\frac{P}{2}$ corresponding to the static deflection a. As a result of the rolling oscillation, the springs on one side will be deflected $a + x$, while those on the other side will be deflected $a - x$. If OE

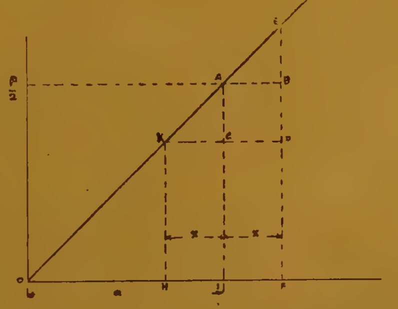

Fig. 3.

is the deflection diagram of the springs (fig. 3), a positive quantity of work will

be absorbed equal to the rectangle AIFE and negative work equal to AKHI, giving a total amount of work absorbed ABCD $= x$BD.

But $$BD = BE = \frac{Px}{2a},$$

therefore the work absorbed is equal to $\frac{Px^2}{2a}$ so that :

$$\frac{P}{2g}(\rho^2 + h^2)\frac{v^2 i^2}{4s^2} = \frac{Px^2}{2a} \quad . . . \quad (11)$$

from which :

$$\frac{x}{a} = \frac{vi}{2s}\sqrt{\frac{\rho^2 + h^2}{ga}} \quad . . . \quad (12)$$

However, $\frac{x}{a}$ is the relative variation of the load on the springs, consequently that of the load on the wheels will be :

$$\Delta = \frac{x}{a} \cdot \frac{m}{s} = \frac{vmi}{2s^2}\sqrt{\frac{\rho^2 + h^2}{ga}} \quad (13)$$

This equation shows that Δ is directly proportional to vi.

Therefore, when the speed of trains on a line is increased, if it is desired to keep Δ unchanged, i must be reduced in the same proportion, and therefore the length of the transition curves proportionately to the speeds of the trains.

This result is in agreement with the conclusions of Messrs. Descubes and Petersen.

With regard to locomotives, it is also necessary to take into consideration the fact that the vertical forces exerted by the crossheads on the slide bars, air resistance and the pull on the drawbar, tend to lift the leading end. For *Pacific* type locomotives, which have a considerable length, the decrease in the load on the bogie springs, due to these causes, is about 6 %.

Having established the above equations, we are in a position to calculate the coefficient of safety against derailment on straight sections of track which have one rail superelevated.

III. *Stability on curves.* — We will now consider the stability of a vehicle on a curve, taking in the first place the case which still exists on certain lines where there is a sudden change from the straight to a circular curve without any transition curve. For this case the variation in the pressure of the wheels on the rails and in the pressure exerted against the outside rail will be calculated. We will then suppose that between the straight and the circular curve a transition curve of a given length is introduced and again determine the variation of the pressure of the wheels upon the rails and the lateral thrust. We shall then be in a position to appreciate the advantages which may be obtained by transition curves and to determine the coefficient of security against derailment under such conditions.

We have already stated that when a vehicle enters or leaves a curve not provided with a transition curve, it is subjected to a rolling oscillation due to the instantaneous application of centrifugal force and a nosing oscillation produced by the sudden change of direction.

Let us first consider the rolling oscillation due to the action of the vehicle which may be calculated by the following formulæ due to Mr. Marié.

Let P be the total weight of the vehicle, which may be divided into spring-borne weight P_1, and non-spring-borne weight P_2 (fig. 4). Let A and B be the points of contact with the rails, α the angle of superelevation of the outer rail, β the angle which the spring-borne portion makes with the plane of the wheels under the action of the weight and the centrifugal force, O the position of the axis of the rolling oscillation [1], G_1 the centre of

(1) According to Mr. Herdner, the axis of rolling oscillation in locomotives is somewhere near the height of the driving axles. According to Mr. Marié, it is a few centimetres higher and is given by the intersection of the centre line through the spring-borne portion and that parallel to the rails through the points of contact of the horn blocks with the axle-

gravity of the spring-borne portion, G_2 that of the non-spring borne portion, C and D the points at which the springs are applied. Let us also call :

k the height of O above the track;

h_1 and h_2 the heights of G_1 and G_2 above the rail level;

Q and R the reactions between wheel and rail perpendicular to the rail;

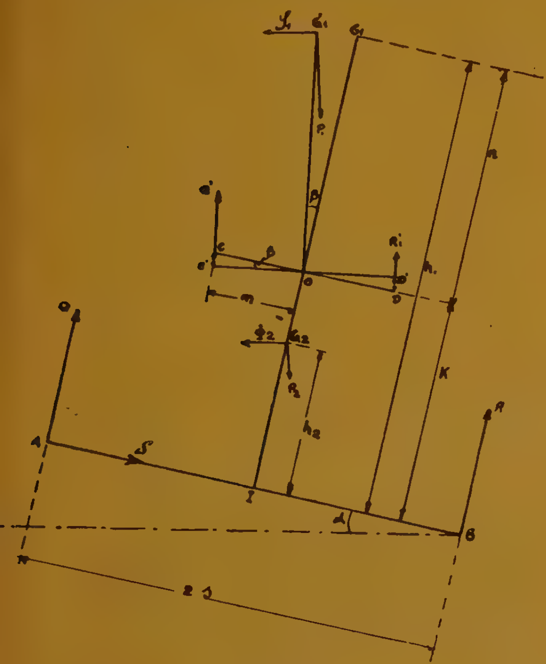

Fig. 4.

S the reaction between the flange of the outside wheel and the rail, Φ_1, Φ_2 the centrifugal forces due to the spring-borne mass and non-spring borne mass at a speed v which is supposed to be constant, and let Φ be their sum;

Δ the relative variation of pressure of the wheels on the inside rail under the action of forces Φ_1, Φ_2 and the superelevation of the outside rail;

R the radius of the circular curve.

The relative variation δ of the load on the springs on the inside of the curve is given by the equation :

$$\delta = \frac{mn}{m^2 - an}\left(\frac{2\Phi_1}{P_1} - \tan\alpha\right) \quad . \; . \quad (14)$$

The angle β of the inclination of the spring-borne portion to the plane of the axles is given by :

$$\tan\beta = \frac{an}{m^2 - an}\left(\frac{2\Phi_1}{P_1} - \tan\alpha\right) \quad (15)$$

The speed v in metres per second corresponding to a given value of Δ may be found from the equation :

$$v^2 = \frac{\Delta\, sg\mathrm{R}\left(1 + \frac{P_2}{P_1}\right)}{2h_1 + h_2\frac{P_2}{P_1} + \frac{2an^2}{m^2 - an}} + \frac{1}{2}g\mathrm{R}\tan\alpha \; (16)$$

whence

$$\Delta = \frac{2h_1 + h_2\frac{P_2}{P_1} + \frac{2an^2}{m^2 - an}}{sg\mathrm{R}\left(1 + \frac{P_2}{P_1}\right)}\left(v^2 - \frac{1}{2}g\mathrm{R}\tan\alpha\right) \; (17)$$

Finally :

$$\mathrm{S} = \Phi_2 + \Phi_1\left(1 + \frac{n^2}{n^2 + \sigma^2}\right) - (P_1 + P_2)\sin\alpha \;\; (18)$$

where σ is the radius of gyration of the spring-borne portion about an axis parallel to the track and passing through its centre of gravity.

Since, for all practical purposes, $\sigma = n$, we may write :

$$\mathrm{S} = \Phi_2 + 1.5\Phi_1 - (P_1 + P_2)\sin\alpha. \quad (19)$$

By equating Δ to unity in equation (16) we can obtain the speed at which the vehicle will turn over.

After entering the curve, the rolling oscillation given by the above equations rapidly dies away as it is damped out by the friction between the axles boxes and horns and by the internal friction of the

boxes. For the amount of superelevation of the outside rail used in railway practice, it may be taken that the position of rolling oscillation is not affected.

springs, so that after a time the spring-borne portion takes up a constant inclination to the plane of the axles, and we then have for the remainder of the circular curve :

$$\delta = \frac{mn}{m^2 - an}\left(\frac{\Phi_1}{P_1} - \tan\alpha\right) \quad . \quad . \quad (20)$$

$$\tan\beta = \frac{an}{m^2 - an}\left(\frac{\Phi_1}{P_1} - \tan\alpha\right) \quad (21)$$

$$v^2 = \frac{\Delta s g R\left(1 + \frac{P_2}{P_1}\right)}{h_1 + h_2\frac{P_2}{P_1} + \frac{an^2}{m^2 - an}} + gR\tan\alpha \quad (22)$$

$$\Delta = \frac{h_1 + h_2\frac{P_2}{P_1} + \frac{an^2}{m^2 - an}}{sgR\left(1 + \frac{P_2}{P_1}\right)}(v^2 - gR\tan\alpha) \quad (23)$$

$$S = \Phi - P\sin\alpha \quad . \quad . \quad . \quad (24)$$

It may be noted that if we have :

$$\tan\alpha = \frac{v^2}{gR},$$

that is to say, if the superelevation of the outside rail is sufficient to counteract the effect of centrifugal force, equation (23) would give $\Delta = o$.

By comparing equation (15) with equation (21), we see that if the curve had no superelevation, the value of the angle β given by equation (15) will be double that given by equation (21). For this reason, Mr. Marié calls the rolling which is produced on curves without transition curves « double amplitude oscillation ». If, on the other hand, we compare the equations (15 to 18) with equations (20 to 24), we immediately see that for the same value of Δ it is necessary to reduce the speed at the entrance and end of curves without transition curves. The same equations show that in order to reduce Δ and increase v, it is necessary to adopt considerable superelevation.

If now we introduce a transition curve between the straight track and circular curve, the double amplitude oscillation will not be produced, and as the radius of curvature decreases the original curve, it follows that the limit of speed given by equation (22) may very reasonably be used for this case.

It should be noted that the values given by equations (14 to 18) represent maxima which are not attained in practice, either because the rolling oscillations are not free oscillations as about a mechanical axis, but are damped out by the friction of the axleboxes in the horns and by the internal friction of the springs, or else because the trains themselves smooth out the track where the straight is joined to a circular curve by slightly deforming it which is sufficient to reduce the rolling oscillation.

We will now consider the nosing oscillation due to the sudden change of direction of the vehicles at the beginning and end of curves not provided with transition curves.

We will calculate the lateral thrust produced by this oscillation in the very simple case of a bogie coach with a distance b between the centres of the bogie centre pins. We shall see that the majority of *Pacific* locomotives, in which the flanges of the first and second pairs of wheels are reduced in thickness, may be taken as similar to a bogie coach as regards this oscillation.

Let A_0BC_0 be the centre line of the track and BC the circular curve to which the straight line is tangential (fig. 5). Suppose that the points A, B represent the position of the bogie pivots immediately before the vehicle enters the curve, and that the points B, C indicate the position of these pivots immediately after entering the curve. Let G be the displacement of the centre of gravity of the vehicle and c its distance from the point B. Let us consider, for simplicity, that the vehicle is rigid in a transverse direction, that is to say, there are no bolsters, that the curve has no superelevation and that the plane of the rails is horizontal and passes through the centre of gravity. These assumptions, which are not conformed to in practice, make the conditions more unfavourable, as we shall show.

The energy expended by the lateral force which is necessary to make the vehicle move from position AB to a position BC should be equal to the kinetic energy created by this movement.

However, we know that this kinetic energy is equivalent to that which the vehicle would have if its whole mass were centred at the centre of gravity, plus the kinetic energy acquired in turning about a vertical axis passing through the centre of gravity.

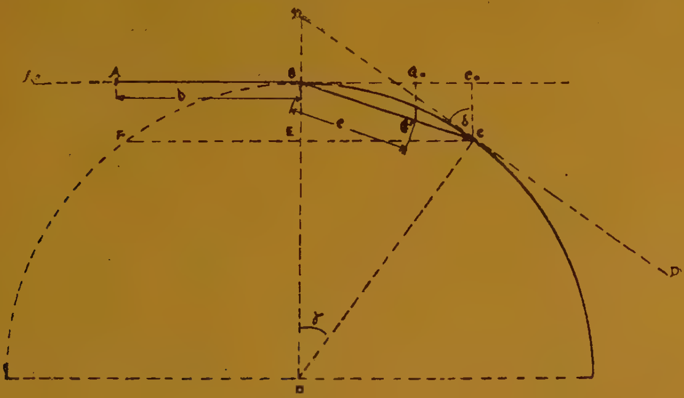

Fig. 5.

The former, the kinetic energy of translation, is obtained by multiplying the mass of the vehicle by half the square of the speed which it has acquired when G_0 has arrived at G. If v is the velocity of the vehicle along the rails, and v' the lateral velocity of the leading bogie pivot when it has reached C, we see from the figure :

$$v' = v \cos \delta = v \tan \gamma = v \frac{b}{R},$$

γ being very small.

Consequently, the speed acquired by the centre of gravity when it has reached G is :

$$v' \frac{c}{b} = v \frac{c}{R},$$

and the kinetic energy of translation :

$$\frac{1}{2} \frac{P}{g} v^2 \frac{c^2}{R^2} \quad . \quad . \quad . \quad . \quad (25)$$

The kinetic energy of rotation is equal to the mass of the vehicle multiplied by half the square of the speed at a radius equal to the radius of gyration about a vertical axis passing through the centre of gravity G. Since the vehicle is passing round the curve at a uniform speed v, a point at the distance equal to the radius of gyration u will turn about G with a speed $\frac{v}{R} u$.

The kinetic energy of rotation will be therefore equal to :

$$\frac{1}{2} \frac{P}{g} v^2 \frac{u^2}{R^2} \quad . \quad . \quad . \quad . \quad (26)$$

and the total kinetic energy is :

$$\frac{1}{2} \frac{P}{g} v^2 \left(\frac{c^2}{R^2} + \frac{u^2}{R^2} \right) \quad . \quad . \quad . \quad (27)$$

This must be equal to the work done by the lateral force F during the time that the pivot C has passed from C_0 to C.

However, as γ is very small, the lateral velocity of the point C is subject to a uniform acceleration, consequently F remains constant while the vehicle passes from a position AB to a position BC. It should be noted that C_0C is equal to the versed sine of an arc of radius R and a

chord approximately equal to $2b$ and that therefore :

$$C_0C = \frac{b^2}{2R}.$$

Therefore :

$$F \cdot \frac{b^2}{2R} = \frac{1}{2}\frac{P}{g}v^2\left(\frac{c^2}{R^2} + \frac{u^2}{R^2}\right)$$

from which :

$$F = \frac{P}{g}\frac{v^2}{R}\left(\frac{c^2}{b^2} + \frac{u^2}{b^2}\right). \quad . \quad . \quad (28)$$

We have stated that by assuming that the force F acts in a horizontal plane which passes through the centre of gravity of the vehicle instead of acting in the plane of the rails, one obtains a greater value for this force than is actually the case. Actually, when a force is exerted upon a body which is partially spring-borne, the spring-borne portions are acted upon less suddenly than the non-spring-borne portions, so that whilst the magnitude of the force is decreased, it acts for a longer time.

This force F must naturally be added to the centrifugal force S.

If now we introduce between the straight and the curve a transition curve of length l, the theory of virtual work shows us that the force F is n times less when the duration of its application is n times greater. But as it increases progressively from o, when $R = \infty$, up to its final value, instead of multiplying F by $\frac{b}{b+l}$, we have to multiply it by $\frac{b}{b+\frac{l}{2}}$.

In this case the approximate expression for the force F acting on the leading bogie at the entrance to the curve is given by :

$$F = \frac{P}{g}\frac{v^2}{R}\left(\frac{c^2}{b^2} + \frac{u^2}{b^2}\right)\frac{b}{b+\frac{l}{2}}. \quad . \quad (29)$$

This equation, which shows us that $F = o$ for $R = \infty$ and for $l = \infty$ demonstrates very clearly the great advantage of transition curves.

In the case of very short vehicles, and where there are no transition curves, it will be found that for curves of the radii used in railway practice, the displacement of the vehicle C_0C becomes very small and about the same as the play which exists between the flanges and the rails, and consequently it is only necessary to take into consideration the kinetic energy of rotation $\frac{1}{2}\frac{P}{g}v^2\frac{u^2}{R^2}$ which, for curves of small radius, may have an appreciable value.

The force F cannot be calculated in this case and depends upon the lateral elasticity of the track and of the underframe. There is a certain analogy existing between the shock which is so produced and that produced by the nosing caused by the play between the flanges and the rails which we have already considered.

It is to be noted that the value of the speed as determined by the preceding equation is directly proportional to B, while that obtained by equations 22, 28 and 29 is directly proportional to $\sqrt{R}$. Consequently, while on main lines with curves of large radius over which bogie vehicles run, to keep Δ constant it is necessary that v should vary as the square root of the radius, on branch lines, with curves of small radius, as a rule without transition curves, over which short vehicles are run, v should vary as the radius.

From what we have shown and taking into account the couple which is due to the longitudinal sliding of the tyres when the radius of a curve is small compared with the coning of the tyres, we may calculate the lateral force which is produced when the vehicle enters a circular curve preceded by a transition curve, and thus obtain the coefficient of security against derailment

In entering or leaving the transition curves, the vehicles are subjected to a rolling oscillation which we have already mentioned ; this is more dangerous when

leaving the curve because it causes a decrease in the load upon the outside rails. In order to overcome this, the outside rail should be given a very small inclination over the inner rail, or it should be prevented by properly laid in curves avoiding the two angular joints in the outer rail at its junction with the straight track, or curves which are perfectly tangential both to the straight line and to the circular curve should be used. Mr. E. Hallade [1] and Messrs. K. Lachmann and R. Rothe [2], among others, have dealt with this question.

The transition curves proposed by these authors are of a sinusoidal form; they have in the first section negative ordinates and should therefore be modified.

Now that we have dealt with the advantages of transition curves from the point of view of safety and easy riding of vehicles on curves, we will return to the fact that at junctions there are curves with no transition curves or superelevation, circumstances being made still worse by the fact that the points being straight cannot be correctly joined up to another line which is also straight. At the beginning and end of these curves, there are consequently rolling oscillations of double amplitude and nosing oscillations as given by formulæ 15 to 18 and 28. To reduce these oscillations it is necessary to use curves of large radius, which may be obtained by having long and flexible points or by deviating both lines instead of one.

There are also curves without transition curves, at places where for special reasons one curve is immediately followed by a reverse curve without any straight or transition curve between the two, and where the track has become out of alignment either as a result of the expansion of the rails in hot weather, or a result of shocks due to nosing oscillations.

In such cases, should synchronism occur, there will be at each point of inflection a rolling oscillation having an amplitude twice that of the preceding oscillation, which would lead to the derailment of the locomotive.

In the same way, the nosing oscillation may, under the circumstances mentioned above, give rise to dangerous synchronism when the axles of the vehicles can be displaced laterally against an elastic resistance without friction.

IV. *Effect of inertia of the non-spring-borne parts of the vehicle and of the gyroscopic action of the wheels on the track.* — Hitherto we have neglected the inertia effect of the non-spring-borne portion, due to the deflection of the rails under the rolling load and to accidental undulations of the track and also the gyroscopic action of the wheels when passing round circular curves and transition curves.

Let us first deal with the inertia effect. We shall see that in actual steam locomotive practice, supposing that the track is in a satisfactory state of maintenance, it is negligible, even for the driving axles.

Mr. Marié has calculated the maximum vertical acceleration of an axle in the case where the rails under the effect of load take up a sinusoidal deformation. This is checked at the points C and E where $x = \frac{\pi}{2}$ and $x = \frac{3\pi}{2}$ (fig. 6). At C we have the maximum negative value and at E the maximum positive value; the former is naturally the more interesting from the point of view of derailment.

The maximum acceleration is given by

$$\gamma = \mp A h \frac{v^2}{l^2} \quad . \quad . \quad . \quad (30)$$

where A is a numerical coefficient equal

(1) E. Hallade : « Nouvelle méthode de raccordement des courbes » (New method of setting out transition curves) (*Revue Générale des Chemins de fer et des Tramways*, 1908, p. 261).

(2) K. Lachmann et R. Rothe : « Zur Konstruktion des Uebergangsbogens für Eisenbahngleise » (Note on the construction of railway transition curves), *Zeitschrift für angewandte Mathematik und Mechanik*, 1922, p. 45.

to about 20, h the deflection of the rail between the points C and E, v the velocity in metres per second and l the length o the rails.

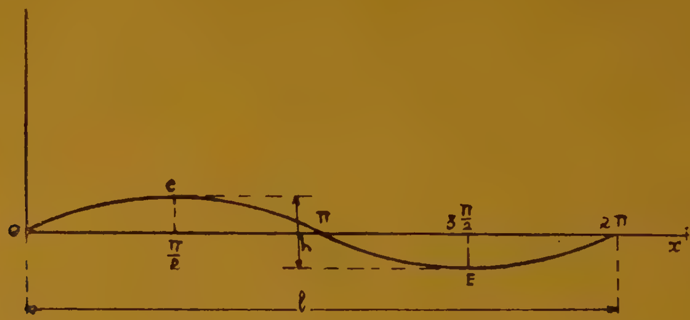

Fig. 6.

Given the deflection of the track under the effect of the wheel loads, we can apply the preceding expression at the point B, that is to say, the middle of the rail where the non-spring-borne weight is at the highest point of its oscillation (fig 7). However, at the joints, the acceleration is greater because of the angle made between the rails at D. It may be determined experimentally by special accelerometers and gives rise to appreciable deflections at the joint.

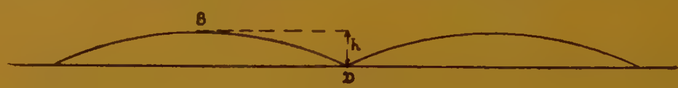

Fig. 7.

The expression given above shows us that to reduce γ it is necessary to reduce h and increase l, or in other words, to keep the track in a good state of repair and use long rails.

If we know γ we can calculate the pressure of the wheels on the rails at the point B. If we designate by p the pressure of the wheels on the rails when the vehicle is at rest, and by P the weight of the wheels and axle, supposing that h is very small compared with the static deflection of the springs, the pressure at the point B will be :

$$R = p - \frac{P}{2g}\gamma \quad \ldots \quad (31)$$

From this it will be seen that it is advisable to fit the leading axles of a locomotive with light wheels, and consequently wheels of small diameter.

The preceding formula can also be applied to the case where the track has horizontal undulations. In this case we have :

$$R' = F \pm \frac{P}{g}\gamma' \quad \ldots \quad (32)$$

by writing

$$\gamma' = 20h\frac{v^2}{l^2}.$$

F is the horizontal thrust exerted on the axle under consideration, if no deviations occur, P the weight of the axle and parts rigidly connected to it, h the deflection and l the length of the sinusoidal deviation. In this case it is the maximum value of the force which we wish to obtain, and consequently the maximum positive value of the acceleration. As the length l is increased, the increase in the pressure decreases as the inverse square of this length.

The foregoing considerations show why electric locomotives should have their motors separated from the axles and be suspended elasticly both in the vertical and horizontal directions.

Let us now consider the gyroscopic action of the wheels due to curves in the track. We know that if a gyroscope rotates about its axis OA with an angular velocity ω, and if the point A is made to rotate slowly with an angular velocity ω' about the point O so that the axis OA moves in a desired plane, a force F will be produced at A perpendicular to the plane in which the axis OA turns, the direction of which is dependent upon the direction of the gyroscope around its axis OA and of the axis OA about the point O (fig. 8).

Fig. 8.

The value of F may be obtained from the equation for the gyroscopic moment, which is given by :

$$Fl = I\omega\omega' \quad . \quad . \quad . \quad . \quad . \quad (33)$$

where l is the distance OA and I the moment of inertia of the gyroscope about the axis OA.

Consider now three successive positions of the axle as it passes around a curve of radius R (fig. 9). To make the axle pass from the position M_1 N_1 to the position M_2 N_2, we may move it parallel to its original direction so that the middle points C_1C_2 coincide, and then turn it about a vertical axis passing through C_2. As the wheels turn about their axle they behave as two gyroscopes, and consequently this rotation which we have just been considering results in forces perpendicular to the plane of rotation of the axle, which forces produce an increased pressure of the wheels on the rail at M and an eqnal decrease in pressure on the rails at N.

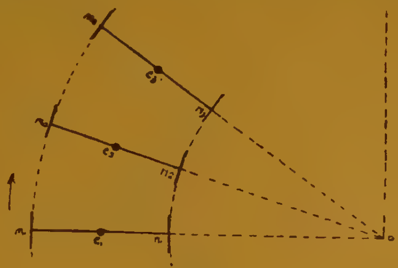

Fig. 9.

This result was obtained by Foucault as long ago as 1878.

If p is the weight of the rim of the wheel, and r the distance of the centre of gravity of the section of the rim from the centre of the wheel, we have as an approximation :

$$I = \frac{p}{g} r^2.$$

If v is the speed of the train and R the radius of the curve, then :

$$\omega = \frac{v}{r}, \quad \omega' = \frac{v}{R}$$

and consequently :

$$Fl = 2 \frac{p}{g} \cdot \frac{r}{R} v^2 \quad . \quad . \quad . \quad (34)$$

where l, the distance between the forces forming the couple, is equal to the gauge.

If the axle carries a non-spring-borne electric motor, it is necessary to substitute for p the total weight of the wheel and of the motor and for r the radius of gyration of the whole.

As we shall see in the examples, this gyroscopic effect is generally negligible, even for very high speeds, when the electric motors or the non-spring-borne portion of these are not mounted on the axles.

Another gyroscopic action of the wheels is that due to the superelevation of the

outside rail above the inside rail on transition curves : each axle of the vehicle in this case turns slowly about a horizontal axis parallel to the track, giving rise to a gyroscopic moment in a horizontal plane and tending to turn the vehicle about a vertical axis so as to decrease the pressure at the front of the vehicle against the outer rail (fig. 10).

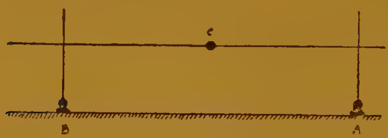

Fig. 10.

The distance between the forces forming the couple is equal to the rigid wheel base of the vehicle. By putting in the formula (33) :

$$I = \frac{p}{g} r^2; \quad \omega = \frac{v}{r}; \quad \omega' = \frac{vi}{2s},$$

i being the inclination of one rail to the other and $2\,s$ the gauge, we have for one axle :

$$Fl = \frac{p}{g} \frac{r}{s} iv^2$$

and for all the axles :

$$Fl = \Sigma \frac{p}{g} \frac{r}{s} iv^2 \quad . \quad . \quad . \quad (35)$$

In the case where a motor is carried on any of the axles, it is necessary to introduce into this formula the modifications that have been mentioned previously.

This action, generally negligible in locomotives, is not so in the case of electric rail cars when the speed is very high : the motors are heavy and turn in the opposite direction to the wheels, and i has a considerable value.

Some of the formulæ we have established assume that the vehicles are under practical conditions, while others assume more unfavourable conditions in order to simplify the problem. Mr. Marié has verified the majority by means of experiments and has found that they give worse results than those obtained in practice; this would be expected, because in all cases the most unfavourable condition, that of synchronism is presumed.

V. *Examples.* — Let us take three locomotives of the *Pacific* type, which we will denote by letters A, B and C, with horizontal cylinders, and having their revolving parts completely balanced, which differ however as regards the springing and bogie side control arrangements. These locomotives, the coupled axles of which carry a load of 18 t. and which are capable of attaining a speed of about 130 km. (80 miles) per hour, give rise to very high stresses in the track.

By applying to these locomotives the calculations established above, we can calculate the influence of the characteristics of the track and of the vehicles, on the stresses imposed on the track. In order to obtain practical results, let us take the distance between the wheels, the diameter of the wheels and the spring-borne and non-spring-borne weight as for the Italian locomotive, class 690 (see figure 11).

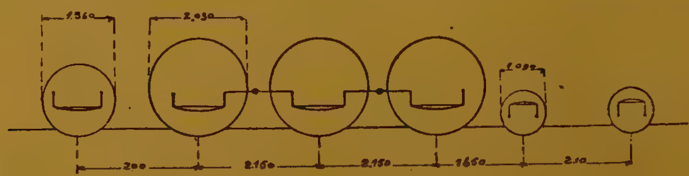

Fig. 11.

Let us also assume that, as in practice, the flanges of the first and second coupled wheels are of reduced thickness, the first rather less than the second, so that this,

which is the driving axle, is not subjected to severe shocks.

The load distribution is as shown in table I.

Let us assume that the bogie frame weighs 2 300 kgr. (5 071 lb.) and the trailing truck frame 900 kgr. (1 984 lb.), and that the springs of the coupled wheels are connected by compensating levers so as to reduce the variation in load due to depressions in the track and to improve the adhesion of the locomotive.

Let us suppose, that engine A is provided with an American type swing link bogie and that the trailing truck has side bearings without lateral control springs; that engine B has an Alsatian type bogie, that is to say, is provided with spherical lateral supports and side control springs and the truck has also spherical lateral supports and side control springs; and finally that the bogie and truck of engine C have a central spherical pivot, and a side control device formed by inclined planes at 15 °/₀ from the horizontal. Engine A is therefore suspended at eight points, engine B at six points and engine C at four points situated at the points of a lozenge shape figure. Let us assume that the springs of engine A have all the same statical deflection of 30 mm. (1 3/16 inches), those of engine B 50 mm. (2 inches), but in engine C the bogie and the truck springs have a statical deflection of 80 mm. (3 5/32 inches) and those of the coupled wheels 50 mm. (2 inches). In this last locomotive, for simplicity of calculation, we will substitute, as regards the pitching oscillations, for the static deflection of the springs of the different axles, one static deflection obtained by taking the geometric mean of the static deflection of the bogie and truck springs and of the coupled wheel springs, taking into consideration the axle loads. This works out at 63 mm. (2 31/64 inches).

As regards the rolling oscillations, as the springs of the coupled wheels are the only ones which effect the stability of the spring-borne masses, it is necessary to substitute for the static deflection of these

TABLE I.

AXLES.		Load on the rails.		Non-spring-borne weight.		Spring-borne weight.	
		in kgr.	in lb.	in kgr.	in lb.	in kgr.	in lb.
Bogie	Leading bogie axle	10 000	22 046	2 700	5 952	7 300	16 094
	Trailing bogie axle	10 000	22 046	2 700	5 952	7 300	16 094
Coupled wheels	1st axle	18 000	39 683	4 200	9 259	13 800	30 424
	2nd axle	18 000	39 683	6 300	13 889	11 700	25 794
	3rd axle	18 000	39 683	4 200	9 259	13 800	30 424
Bissel truck		13 200	29 101	2 800	6 173	10 400	22 928

springs the derived deflection a' given by : $a' = \frac{a}{1 - d}$ where d is the ratio of the spring-borne weight carried by the bogie and truck to the total weight.

After having made the necessary substitutions, we obtain $a' = 83$ mm. (3 17/64 inches).

For engines A and B, assuming that the flexibility of the springs is inversely proportional to the loads, it will be found that the elastic centre and the centre of gravity of the spring-borne part are situated on the same vertical axis, 44 cm. (1 ft. 5 5/16 in.) in front of the second coupled axle; in engine C the vertical through the elastic centre is 27 cm. (10 5/8 inches) in front of that through the centre of gravity of the spring-borne portion; in the three locomotives the vertical through the centre of gravity of the whole engine is 48 cm. (1 ft. 6 7/8 in.) in front of the second coupled axle.

Let us now consider the control arrangements.

The resistance to lateral displacement of the bogie of engine A is equal to P tan α where P is the weight on the bogie and α the angle of inclination of the swing links to the vertical. Supposing that the length of these is 15 cm. (5 29/32 inches) and their maximum lateral displacement is 5 cm. (2 inches) so as to allow the engine to pass round curves of 200 m. (10 chains) radius, this force will have a maximum value of 4 100 kgr. (9 039 lb). the weight P being 12 300 kgr. (27.117 lb.).

To this we must add the frictional resistance between the hole in the swing link and the pin which, as there are two pins, is given by :

$$2 P \frac{r \sin \varphi}{l}$$

where r is the radius of the pin, l the length of the swing links and $\sin \varphi = \tan \varphi$ which is equal to the coefficient of friction between the hole and the pin.

If $r = 3.5$ cm. (1 3/8 inches) and $\tan \varphi = 0.10$, the resistance is as much as 580 kgr. (1 279 lb.). The initial resistance to displacement is therefore 580 kgr. and the maximum resistance 4 680 kgr. (10 317 lb.) (fig. 12).

As the weight carried by the trailing truck is 9 500 kgr. (20 944 lb.), taking the same coefficient of friction as in the preceding case, since the surfaces in contact are supposed to be reasonably lubricated, the resistance to lateral displacement is 950 kgr. (2 094 lb.), the trailing truck not being provided with a lateral control spring.

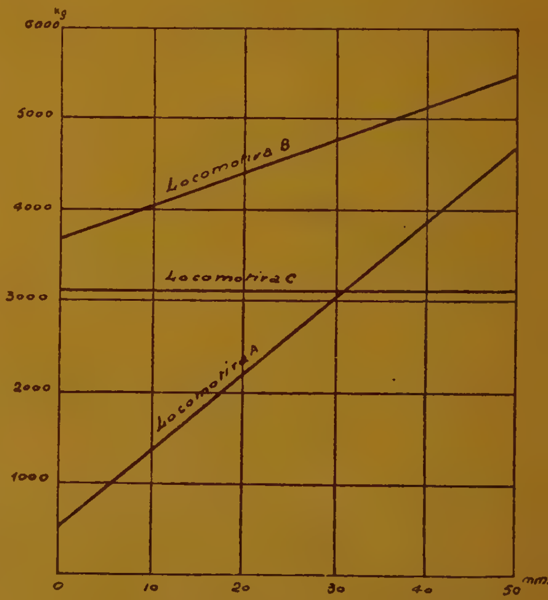

Fig. 12.

For the bogie of engine B, if the lateral control spring has an initial tension of 2 500 kgr. (5 512 lb.), and deflects through 30 mm. (1 3/16 inches), under a load of 1 000 kgr. (2 205 lb.), taking into account the frictional resistance due to the load carried by the bogie, we have an initial resistance to lateral displacement of 3 730 kgr. (8 223 lb.) and a maximum resistance of 5 400 kgr. (11 905 lb.) (fig. 12). As the trailing truck is supposed to be fitted

with a lateral control spring having an initial tension of 1 500 kgr. (3 307 lb.) and a flexibility equal to that of the bogie spring, we have a minimum resistance of 2 450 kgr. (5 401 lb.) and a maximum of 4 120 kgr. (9 083 lb.).

For engine C, under the assumed conditions, the resistance to lateral displacement of the bogie and of the truck are constant and their respective values are 3 075 and 2 380 kgr. (6 779 and 5 247 lb.) (fig. 12).

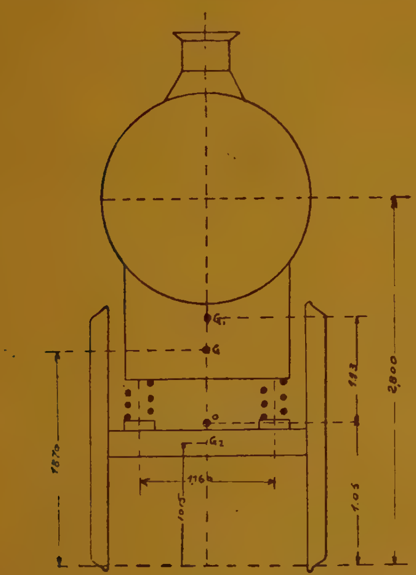

Fig. 13.

Let us now consider the cross section of the locomotive. Figure 13 gives us the dimensions which are necessary in the calculations: suppose, moreover, that the radius of gyration about a horizontal axis parallel to the track and passing through the centre of gravity of the locomotive is equal to 78 cm. (2 ft. 6 3/4 in.) and that that about a vertical axis also passing through the centre of gravity is 2.20 m. (7 ft 2 5/8 in.).

Stability of the engines on straight track. — Let us first calculate the maximum periodic deflection of the track at the rail joints for the case when these synchronise with the oscillations at the critical speed, and then verify the stability of the engines on straight track when the rails are at the same level and when one rail is superelevated so that we can take into account the influence of inclination of one rail relatively to the other on the coefficient of safety.

To determine this coefficient, we will calculate the ratio $\frac{F}{\Pi} = A$ between the maximum lateral pressure of the axle under consideration and the minimum load of the wheel upon the rail, and we will then establish from formula (2) the ratio $\frac{F}{\Pi} = B$ which must be satisfied for the axle to become derailed; the ratio $C = \frac{B}{A}$ gives us the coefficient of security against derailment.

In order that we may take the most unfavourable conditions, we will suppose that during a nosing movement synchronism occurs, and that at the moment that the wheel strikes the rail the load on the wheel is partially removed as the result of a local depression of 2 cm. (25/32 inch) in the track. We must also take into account the effect which the forces set up by the action of the steam, air resistance and tractive effort at the drawbar have on the leading part of the locomotive.

For engine A, in which the bogie has a low initial resistance to lateral displacement, we will consider the stability of the leading coupled axle; for engines B and C, in which the bogie offers a considerable resistance to lateral displacement, we will consider the stability of the leading bogie axle under the assumption that the pressure is divided equally between its two axles.

In applying the formulæ (6), (8) and (9), we must not loose sight of the fact that the bogie of engine B acts as a longitudinal compensating beam, while that of engine C acts both longitudinally and transversly

TABLE II.

Locomotive.	Track.	Axles.	h m. (inch).	Δ	F kgr. (lb.).	Π kgr. (lb.).	A	B	C
Engine A.	On straight line	... Leading coupled axle.	... 0.008 (5/16)	0.283	3 400 (7 496)	7 125 (15 708)	0.48	1.14	2.37
	On straight line with one rail superelevated $i = 0.003$ . . .	...	...	0.376	3 400 (7 496)	6 455 (14 231)	0.53	1.14	2.15
Engine B.	On straight line	... Leading bogie axle.	... 0.013 (33/64)	0.333	2 265 (4 993)	3 820 (8 422)	0.59	1.14	1.93
	On straight line with one rail superelevated $i = 0.003$ . . .	...	...	0.403	2 265 (4 993)	3 524 (7 769)	0.64	1.14	1.78
Engine C.	On straight line	... Leading bogie axle.	... 0.016 (5/8)	0.282	1 538 (3 391)	3 970 (8 752)	0.39	1.14	2.92
	On straight line with one rail superelevated $i = 0.003$ . . .	...	...	0.308	1 538 (3 391)	3 872 (8 536)	0.40	1.14	2.85

and that the trailing bogie of this engine is provided with a central spherical pivot.

Therefore, by putting in the formulæ (2) to (9), $\beta = 60°$, $\varphi = 0.20$, $\varphi' = 0.10$, $K = 0.30$, $\varepsilon = 0.025$, $f = 0.063$, $2m. = 1.16$, $2s = 1.50$, and designating by h the maximum periodic deflection of the track for which the condition of synchronous oscillations is satisfied and by Δ the relative decrease of the pressure of the wheels on the rails, we obtain the results shown in table II.

This table shows that engines B and C may run over tracks which are in a much worse state of repair than locomotive A without the pitching oscillations becoming excessive. As regards lateral effects on the track, engine C is the best; engine A exerts a much greater force than the other two, the pressure being produced by the leading coupled axle.

The inclination of the outside rail above the inside rail has a very small effect on the wheel loads of engine C, which is flexible in a longitudinal direction, and has an appreciable effect on the wheels of engines A and B, which lack this flexibility.

As regards engines B and C, it may be well to note that by applying formula (3) we find that $d = 30$ mm. (1 3/16 inches). The play between the flanges and the rails being 25 mm. (63/64 inch), this is to say that a small part of a shock, which we cannot calculate theoretically, but which may be determined experimentally, is resisted by the leading coupled wheel. The reason for this is that the frictional resistance of the bogie centreing gear is not sufficient to completely damp out the shock. This disadvantage could be overcome by increasing by a certain amount the weight carried by the bogie, which would thus increase the coefficient of safety against derailment. The lateral oscillations would also be appreciably decreased by decreasing the play between the flanges and the rails to 20 mm. (25/32 inch).

Stability on curves. — Stability against

overturning and speed limits. — Supposing that the superelevation of the outside rail on curves of a constant radius is 75 mm. (2 61/64 inches), a usual figure for lines run over by fast trains, then $\tan \alpha = 0.05$. Let us assume the condition that the decrease of the pressure of the wheels on the inside of the curve and the increase of the pressure on the outside rails should not exceed 30 %; we have then a coefficient of safety against turning over equal to 3.33.

Given the condition : $\Delta = 0.30$ = a constant the formulæ (22) and (29) which give the rolling oscillations on a curve and the nosing oscillations when entering the curve show us that the speed ought to vary as the square root of the radius of the curve, and that consequently the centrifugal force Φ and the lateral pressure S are independent of the speed.

Under this assumption, we have calculated and show in table III the limiting speeds which may be attained by engines A, B and C for curves of radii from 2 000 m. to 200 m. (100 to 10 chains).

TABLE III.

RADIUS OF CURVE.		Limiting speed per hour.					
		Engine A : $a = 0.03$.		Engine B : $a = 0.05$.		Engine C : $a = 0.083$.	
Metres.	Chains.	Kilometres.	Miles.	Kilometres.	Miles.	Kilometres.	Miles.
2 000	100	206	128.0	203	126.1	198	123.0
1 [illegible]00	90	195	121.2	192	119.3	187	116.2
1 600	80	184	114.3	181	112.5	177	110.0
1 400	70	172	106.9	169	105.0	165	102.5
1 200	60	159	98.8	157	97.6	153	95.1
1 000	50	145	90.1	143	88.9	140	87.0
800	40	130	80.8	128	79.5	125	77.7
600	30	113	70.2	111	69.0	108	67.1
500	25	103	64.0	101	62.8	99	61.5
400	20	92 (82)	57.2 (51.0)	90 (81)	55.9 (50.3)	88 (79)	54.7 (49.1)
300	15	79 (62)	49.1 (38.5)	78 (61)	48.5 (37.9)	76 (59)	47.2 (36 7)
200	10	65 (41)	40.4 (25.5	64 (41)	39.8 (25.5)	62 (39)	38.5 (24.2)
$\frac{\Phi}{P}$		0.166		0.163		0.154	
$\frac{S}{P}$		0.116		0.113		0.104	

It will be seen from this table that the limiting speeds decrease slightly with the increase of the static deflection of the springs. The numbers in brackets represent the limiting speeds which should be attained on lines where no transition curves are used and which are run over by vehicles of short wheel base. In accordance with formula (26), these speeds are proportional to the radii of the curves.

The speeds above 130 km (80 miles) per hour which appear in the table are purely theoretical, either because the steam locomotives have not sufficient power to haul

trains at such speeds, or because the track would not be sufficiently strong to withstand the inertia action of the non-spring-borne weights.

Stability on curves of constant radius. — Distribution of lateral forces (fig. 14). — By means of the above table we can easily calculate the lateral force S applied at the centre of gravity of the locomotive. Let us designate by X, Y, Z and U the reactions of the outside rail on the bogie, the first and third coupled axle and the trailing truck. Assuming that the flange of the second pair of coupled wheels are of reduced thickness, it can be taken that they do not exert any appreciable pressure against the outer rail, which is an advantage, because this axle is cranked and is therefore weaker than the others.

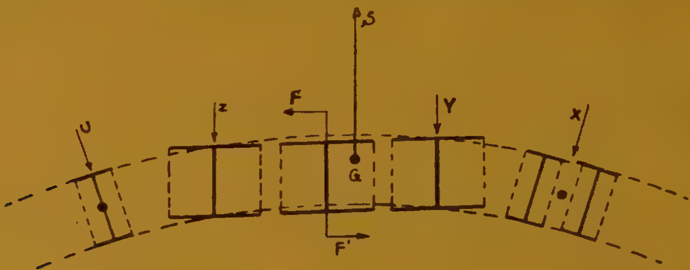

Fig. 14.

Let us assume that the engines are running round a curve of 800 m. (40 chains) and, for the calculation of the unknown terms, let us suppose that the frictional forces are overcome by the vibrations.

In this cases the forces X, U can be easily determined for engines A and B, when one has calculated the lateral displacement of the bogie and truck (fig. 12). By applying formula : $e = \frac{l^2}{8R}$ where e is the displacement required and l is twice the distance of the axis of the bogie or truck from the centre of gravity, it will be found that $e = 12$ mm. (15/32 inch) for the bogie and $e = 13$ mm. (33/64 inch) for the truck. For engine C the values X and U are known and independent of the displacement as mentioned above.

As regards the other unknown terms : Y, and Z, these must be deduced from the equations of equilibrium of translation and rotation about the centre G. It may be remembered, however, that when the radius of the curve is less than 1 200 m. (60 chains), the coning of the tyres of the driving wheels which have a diameter of 2.030 m. (6 ft. 8 in.) is insufficient to avoid slipping of the outside wheels relatively to the inside wheels. Therefore, if P is the weight on the driving wheels, there will be a couple, the moment of which is given by $P \cdot \varphi\, 2s = 8\,100$ Kg.-M. (58 600 foot-pounds).

This negative moment has an influence on the value of the unknown terms and consequently effects X for engines A and B, and U for engine B. However, as a small lateral displacement of the centreing gear on these engines does not appreciably effect the pressure of the wheel against the rail, we may assume that X and U are unchanged and that we may directly calculate Y and Z. We have then calculated for the three locomotives the stability of the leading coupled axle and for engines B and C that of the leading bogie wheel when the radius of the curves is greater than 550 m. (27 1/2 chains), that is to say, when there is no slipping of the outside wheels relatively to the inside wheels, and when the radius is less than 550 m., that is to say, when such slipping occurs. The couple which acts

on the bogie is in this case 3 000 Kg.-M. (21 700 foot-pounds); the wheel base of the bogie being 2.10 m. (6 ft. 11 in.), this couple increases the reaction of the leading wheel by 1 428 kgr. (3 148 lb.).

In making this calculation of the stability, we will suppose that the axle under consideration is subjected to a decrease in the load due to an isolated deflection of 2 cm. (25/32 inch.) of the track, and that the outside wheel is subjected to an increase of pressure of 30 %. For the bogie wheels we will also take into account the decrease in the load due to the action of the steam.

Taking into the account the effect of the longitudinal compensating beams on the variation in the load on the axles which they connect, we obtain the results shown in table IV.

The results obtained show that in order to obtain satisfactory distribution of the lateral forces, one should use bogies and trucks which have a high initial resistance to lateral displacement and bogies which act as longitudinal compensating beams. It is also advisable to give sufficient coning to the tyres of bogie wheels and to use small wheels so as to decrease the slipping of the outside wheels relatively to the inside wheels on curves of small radius. Moreover, small wheels have less tendency to derail than large wheels, and as they are much lighter, they exert much smaller inertia effects on the track.

The coefficient of safety against derailment of the leading bogie axle of engine A is low for radii of 300 m. (15 chains). It should be noted that for lines run over by fast trains, the radii of curves is generally greater than 500 m. (25 chains), so that for the bogie of this locomotive there is a very good coefficient of safety against derailment. Further, as we have already stated, it is only necessary to increase the flexibility of the springs in order to appreciably improve the stability of the bogie. Moreover, the coefficient of safety of the leading coupled axle of this locomotive, for curves of large radius, is rather low; for the smaller radii it is better, since as X decreases, F consequently decreases.

The results which we have obtained draw our attention to another fact, that is, that the weak point of present day locomotives is the lateral rigidity of the coupled axles. On certain electric locomotives this drawback can be overcome by fitting the axle-boxes of driving wheels with inclined planes. Unfortunately in steam locomotives it is difficult to do this owing to the presence of the connecting and coupling rods. However, a partial lateral elasticity is obtained with excellent results, by means of the Krauss-Helmholtz, Italian, Zara, and other bogies, in which the leading carrying wheel is connected to the leading coupled wheel, which can move in a transverse direction.

Stability on transition curves. — We have pointed out that the *Pacific* type locomotive, in which the flanges of the first and second coupled axles are of reduced thickness, behaves on a transition curve in an almost similar manner to a bogie coach. The separation of the lateral frictional resistances of these two axles places us under more unfavourable conditions than occur in actual practice.

Let us apply to our typical locomotives formula (29) making l = 40 m. (43.7 yards) that is to say, adopting a transition curve of the minimum length in accordance with the permanent way instructions issued by the Italian railways. We shall have :

Engine	A	F = 900 kgr.	(1 984 lb.)
—	B	F = 890 —	(1 962 lb.)
—	C	F = 840 —	(1 852 lb.)

It should be noted that while the force F gradually increases from zero to the final value, the friction set up in the bogie centreing gear is not overcome by the vibrations. It is equal to 580 kgr. (1 279 lb.) for engine A and 1 230 kgr. (2 712 lb.) for engines B and C, and therefore gives rise to an increase in the lateral force at the instant when the engine enters upon a circular curve. For engines B and C, this increase

TABLE IV.

	Type of locomotive.	Axle.	Radius of curve, in metres (in chains).	F kgr. (lb.).	Π kgr. (lb.).	A	B	C
A	X = 1 020 kgr. (2 249 lb.).	Leading bogie axle. .	> 550 (27 1/2)	510 (1 124)	2 745 (6 052)	0.19	1.14	6.0
	Y = 6 430 kgr. (14 176 lb.).	— —	= 300 (15)	2 678 (5 904)	2 745 (6 052)	0.97	1.14	1.18
	Z = 2 650 kgr. (5 842 lb.).	Leading coupled axle.	= 800 (40)	6 430 (14 176)	9 140 (20 150)	0.705	1.14	1.61
	U = 0 kgr. (0 lb.).							
	S = 10 100 kgr. (22 267 lb.).							
B	X = 2 900 kgr. (6 393 lb.).	Leading bogie axle. .	> 550 (27 1/2)	1 450 (3 197)	4 970 (10 957)	0.292	1.14	3.90
	Y = 4 045 kgr. (8 918 lb.).	— —	< 550 (27 1/2)	2 878 (6 345)	4 970 (10 957)	0.58	1.14	1.96
	Z = 1 050 kgr. (2 315 lb.).	Leading coupled axle.	< 1 200 (60)	4 045 (8 918)	9 945 (21 925)	0.407	1.14	2.80
	U = 1 835 kgr. (4 045 lb.).							
	S = 9 830 kgr. (21 671 lb.).							
C	X = 1 845 kgr. (4 067 lb.).	Leading bogie axle. .	> 550 (27 1/2)	923 (2 035)	5 160 (11 376)	0.179	1.14	6.37
	Y = 5 117 kgr. (11 281 lb.).	— —	< 550 (27 1/2)	2 351 (5 183)	5 160 (11 376)	0.46	1.14	2.47
	Z = 683 kgr. (1 506 lb.).	Leading coupled axle.	< 1 200 (60)	5 117 (11 281)	10 160 (22 399)	0.505	1.14	2.25
	U = 1 425 kgr. (3 142 lb.).							
	S = 9 070 kgr. (19 996 lb.).							

TABLE V.

Description of locomotive.	Axle under consideration.	Radius of curve, in metres (in chains).	F kgr. (lb.).	Π kgr. (lb.).	A	B	C
Engine A	Leading axle of bogie.	> 550 (27 1/2)	960 (2 116)	2 745 (6 052)	0.35	1.14	3.26
	— —	= 300 (15)	3 128 (6 896)	2 745 (6 052)	1.14	1.14	1.0
Engine B	— —	> 550 (27 1/2)	2 065 (4 553)	4 970 (10 957)	0.41	1.14	2.77
	— —	< 550 (27 1/2)	3 493 (7 701)	4 970 (10 957)	0.70	1.14	1.62
Engine C	— —	> 550 (27 1/2)	1 538 (3 391)	5 160 (11 376)	0.30	1.14	3.80
	— —	< 550 (27 1/2)	2 966 (6 539)	5 160 (11 376)	0.57	1.14	1.99

is greater than F, that is to say, more than sufficient to produce the rotation of the engine. For engine A it is smaller, and it therefore causes a slight increase in the lateral displacement of the bogie, which produces the 320 kgr. (705 lb.) of lateral pressure necessary to cause the rotation of the engine.

This being so, let us again calculate the coefficient of safety against derailment for the leading axle of the bogie (see table V).

As might be anticipated, the coefficients of safety against derailment at the instant when the engine enters a curve are smaller than while the engine is fully on the curve. This effect is, however, only instantaneous; moreover, should one of the wheels tend to lift, this tendency will be opposed by the increased load carried by its particular spring.

As regards the coefficient of safety against derailment for the first coupled axle, it may be taken as being practically the same as when the engine is entirely on the curve. In the same way the coefficient of safety against overturning is very slightly affected.

As the length of the transition curve increases, the reaction necessary to produce the rotation of the engine decreases.

Rolling oscillation due to the superelevation of the outside rail in transition curves. — By applying the formula (13) and taking $v = 34.7$ m. (113 ft. 10 1/8 in.) per second = 125 km. (78 miles) per hour, we obtain the following percentage variations in the wheel loads :

Engine A	20.2
— B	15.7
— C	12.2

Engine C, the springs of which are more flexible, behaves better than the others. This oscillation is undesirable when leaving a curve, because it has the effect of decreasing the load on the outside wheels and consequently decreases the value of π in the formula for derailment.

Inertia effect of the non-spring-borne masses. — Vertical forces. — If we assume that we have an exceptional deflection of the track of 2 cm. (25/32 inch), that the rails are 12 m. (39 ft. 4 1/2 in.) long and that the speed is 125 km. (78 miles) per hour, formula (30) gives us $\gamma = -3.35$ m. per second per second. As $\frac{\gamma}{g} = -0.341$, we find that the inertia force is equal to 34 % of the weight of the axle under consideration. If this is the leading coupled axle, this gives a decrease of pressure of 1 415 kgr. (3 120 lb.), equal to 8 % of the total weight which it carries. This is not of great importance, but for speeds considerably higher than 125 km. (78 miles) per hour, these inertia forces become very important and make it impossible for locomotives to run at very high speeds, unless we could use a metal of the same strength as stee but lighter for the wheels and axles, and this we are not able to do in the present state of metallurgical knowledge.

With electric traction, as we can use driving wheels of a diameter of about one metre (3 ft. 3 3/8 in.), which are therefore much lighter than the coupled wheels of fast steam locomotives, it is possible to considerably reduce these inertia forces, provided that the motors are completely spring-borne.

Horizontal forces. — By calculating the acceleration γ' which enters into equation (32) and by supposing that the maximum value of the sinusoidal deviations of the track at the joints is 2 cm. (25/32 inch), we find that the leading coupled axle exerts a horizontal inertia force of 1 435 kgr. (3 165 lb.). If we take this into account for motion round a curve, the coefficient of safety against derailment for this coupled axle becomes :

For engine	A . . .	C = 1.32
—	B . . .	C = 2.07
—	C . . .	C = 2.01

Coefficient C is also decreased for the leading bogie axle, since we have to take into consideration an increase of 1 310 kgr. (2 888 lb.) in the pressure of the outside wheels against the rail, the laterally rigid weight of the bogie being 7 700 kgr. (16 976 lb). It might be pointed out, however, that even on poorly maintained railways, we never find deviations of 2 cm (25/32 inch), at the joints.

When a locomotive reaches the points of a turn-out it is subjected to a small instantaneous deviation due to the points; flexible points facilitate the entry of the engine on to the deviated track. If the horizontal disturbances which are produced by the change of direction are sufficient to compromise the stability of the engine, suitable speed reductions are imposed.

Gyroscopic action due to curves and to the superelevation of the outside rail in transition curves. — The former is of no interest, except for coupled axles. By putting in formula (34) $2p = 2\,000$ kgr., $r = 0.95$ m., $R = 800$ m., $v = 33$ m. per second, we find that the variation of pressure on one wheel is only 88 kgr. (194 lb.), that is to say, it may be neglected.

The latter gives rise to a decrease in pressure of each bogie wheel against the outside rail equal to 98 kgr (216 lb.), when i in formula (35) is taken as 0.003. The changes of gradient along the line do not give rise to any difficulty, provided that care is taken to join these up by means of curves of large radius.

In our investigation, as we have supposed that the turning masses are balanced, we have not taken into account oscillation set up by reciprocating masses, because at high speeds these have a much shorter period than the nosing, pitching rolling oscillations, and are easily damped out by the internal friction of the springs.

By following a similar method to that which we have adopted for locomotives, we can easily verify the degree of stability for other vehicles which are used in the make-up of the trains.

Of these vehicles, those most subject to derailment are tenders, luggage vans and four-wheel wagons. In these cases there is a great difference between the fully loaded weight and the weight when empty, so that the flexibility of the springs is insufficient when the vehicle is empty, and also the condition of synchronisation of the oscillations produced by the deflection of the track is not satisfied, as may be easily seen by applying the formula $h \leqslant 4fa$. It is true that these vehicles do not as a rule run at a critical speed; however, a derailment may be produced when a lateral shock occurs simultaneously with a deflection of the track.

VI — Up to the present we have been concerned with the degree of security of vehicles against derailment and overturning on a particular track and for speeds which vary as the square root of the radii. It may also be as well to consider the question of easy riding which is necessary both for the comfort of the passengers and to reduce damage to the rolling stock and to certain classes of goods.

In order that easy riding may be obtained, it is necessary that the accelerations produced by the oscillations of the spring-borne portion of the vehicle shall be small and that they shall quickly die out

Generally speaking, when the degree of stability of the vehicle is high, it is sufficiently easy riding. However, there are certain oscillations which are prejudicial to the stability of the vehicle which have no influence on its easy riding, such as variations in the vertical and lateral pressures of the wheels on the rails produced by the inertia of the non-spring-borne masses and by the gyroscopic action of the turning masses.

We will now consider the vertical oscillations due to deflections in the track and the lateral oscillations at the entrance of circular curves.

Let us suppose in the first place that we have a vertical oscillation having a total amplitude e. We know that the period of the oscillation is given by :

$$t = \pi \sqrt{\frac{a}{g}}$$

Let us assume, as an approximation, that this oscillation may be taken as a uniformly accelerated motion, calling γ the acceleration, we have :

$$\frac{e}{2} = \frac{1}{2}\gamma\left(\frac{t}{2}\right)^2$$

from which :

$$\gamma = \frac{4g}{\pi^2}\frac{e}{a} = \frac{4e}{a} \text{ (approximately).}$$

If we make the usual assumption that under the effect of the wheel loads there are isolated deflections in the track of 2 cm. (25/32 inch), since Mr. Marié has shown (1) that the amplitude of oscillation of the spring-borne masses cannot exceed this deflection, we therefore have :

For engine A : γ = 2.66 m. (8.727 feet per sec. per sec.
— B : γ = 1.60 m (5.249 feet) per sec. per sec.
— C : γ = 1.34 m (4.396 feet) per sec per sec.

and the riding of the vehicle will not be quite satisfactory.

On the other hand, if we consider a bogie coach for which a = 0.30 m. (11 13/16 inches) will work out as 0.27 m. (10 5/8 inches) per second per second, which corresponds to easy riding.

We will now pass on to the lateral oscillations which are produced when the vehicles enter circular curves preceded by transition curves. As we know the total pressure S, we can easily determine the lateral acceleration of the centre of gravity of the spring-borne portion of our locomotives. The maximum value occurs for engine A, which can run at a speed of

(1) « Dénivellations de la voie et oscillations des véhicules de chemins de fer » (Depressions in the track and oscillations of railway vehicles) (*Annales des Mines*, May 1911).

130 km. (80.8 miles) per hour, and is 1.14 m. (3.74 feet) per second per second. As the centre of gravity of the spring-borne portion is at a distance of 1.13 m. (3.707 feet) from the axis of rolling oscillation, while the end of the footplate is 1.45 m. (4.757 feet) from this axis, we shall have at this point an acceleration of 1.46 m. (4.79 feet) per second per second.

Accelerations of a much greater value are obtained at the entrance to curves which have no transition curves and during nosing oscillations when the vehicle is laterally rigid. Mr. Marié has measured accelerations which exceed 2.5 m. (8.20 feet) per second per second. It is for this reason that it is advantageous to give the vehicles sufficient lateral elasticity, both as regards security against derailment and also to obtain easy riding. We have already mentioned the arrangements which are adopted on steam and electric locomotives to give the axles sufficient lateral elasticity. Mr. Marié recommends the use on bogie coaches, intended to run over lines which have not transition curves, of horizontal laminated springs, which, on account of their elasticity and internal frictional resistance, damp out the shocks at the entrance and exit of curves.

The considerations developed above show that the stability and easy riding of a locomotive depend not only on the track but also on the constructional details of the engine itself.

As regards the track, care should be taken that the deformations at the rail joints shall not be greater than those which occur at other points, that the play between the flanges and the rails shall not be excessive and that the superelevation of the outside rail shall take place gradually in transition curves.

As regards the locomotive itself, it is good practice to provide the carrying axles with wheels of a small diameter and with springs of considerable flexibility, to use a bogie centreing control which gives a high initial resistance to lateral displacement, to give to at least some of the driving axles an elastic lateral displacement and to allow the locomotive as a whole a suitable flexibility in a longitudinal direction.

The coefficients of safety against derailment which have been determined are probably lower than is actually the case, since we have assumed that the various oscillations occur simultaneously and that the condition of synchronism occurs between these, whereas it is very unlikely that this would be so in practice.

It would, however, be desirable to check the results obtained by means of direct experiment. There is no lack of the means to do this.

In this connection, a Committee nominated by the « American Society of Civil Engineers » to carry out experiments on the deflections of the track produced by certain locomotives, two being of the *Pacific* type, has used recording apparatus giving the deflections of the rails. It is much to be regretted that the very interesting report of this Committee (1) only contains a small part of the data necessary for the verification of the stability of the engines tested. If all this information had been given, it would have been possible, by comparing the calculated with the experimental results, to have formed an idea of the degree of approximation of Mr. Marié's formulæ.

On straight track, for the two *Pacific* type engines, the American Committee found that when the speed increased from 7.6 km. to 91.5 km. (4.7 to 56.9 miles) per hour, the deflections produced by the coupled axles increased from 20 to 27 %. If we take into account for our engines the maximum variation of the pressure of the wheels produced by the track deflections of 2 cm. (25/32 inch.) and by the vertical inertia forces of the wheels and axles, we obtain for the coupled axles the following increases in deflection :

Engine A. . . . . 36 %

(1) See « Stresses in Railroad Track » in the *Proceedings of the American Society of Civil Engineers*, March 1923.

Engine B. . . . . 28 %
— C. . . . . 25 %

As regards the horizontal effects caused by nosing, the report of the Committee states that these in many cases are 14 % of the vertical effects.

For our part, taking the most unfavourable case, where synchronism occurs, we have found, for the leading coupled axle of engine A, a lateral effect equal to about 16 % of the vertical effect, and for the bogie wheels of engine B and C, lateral effects equal to about 19 % and 13 % respectively of the vertical effects.

We are of course making approximate comparisons, since as regards the engines tested in America we do not know either the details of the bogie or the proportion of spring-borne weight to non-spring-borne weight, nor the flexibility of the springs nor the deflections of the track under the effect of the rolling loads.

As regards the passage round a curve, it is impossible to make any comparison because the wheel base of the engines tested are different from those which we have adopted, and moreover the tests were carried out in a very different manner to that which we have been considering. We will merely mention that, for curves of less than 250 m. (12 1/2 chains) radius with a small amount of superelevation and limited speeds, it was found that the inside rail carried a load considerably more than would be given by the calculations. Moreover, the distribution of load in a longitudinal direction is appreciably different from that which exists on straight track, from which it would appear that under these conditions the compensating levers are not so effective in fulfilling their purpose of reducing the variations of the axle loads. It may therefore be of interest to investigate the reasons for this unequal loading and the suitable means to overcome this disadvantage.

In April 1925, the American Committee published a further report (1) dealing, among other things, with the effects on the track produced by electric locomotives for passenger and freight trains.

The spring-borne portion of these locomotives was carried on symmetrically arranged bogies. The power transmission to the axles was by gearing; in some cases the armatures of the motors were keyed on the axles. Consequently, there were no coupling rods or connecting rods, and there were no effects due to the imperfect balancing of the revolving parts. The locomotives had a certain degree of lateral flexibility, obtained either by means of special springing arrangements or by means of bogies placed, not only at the ends, but near the centre of the locomotives.

On a straight track the relative increases in the vertical loads and the lateral pressures on the track under the effect of speed and of nosing movement were, in general, less than with steam locomotives, either because of the reduced non-spring-borne weight on the driving wheels, or as the result of the absence of unbalanced revolving and reciprocating masses and the greater lateral flexibility of the electric locomotives compared with the steam locomotives.

On curves, the sum and the distribution of the forces were different in accordance with the weight, wheel base and wheel arrangement of the various locomotives tested, and according to their lateral flexibility. The results of these tests were of very great use in the improvement of the factors which influence the running of electric locomotives on curves.

During the tests, it was found that the distribution of the axle loads and the wheel spacing had a considerable infuence on the vertical forces imposed on the track, and the advantage which was obtained at high speeds by transition curves, especially when the non-spring-borne mass was considerable was clearly shown.

In addition to the devices used by the

(1) See « Stresses in Railroad Track » in the *Proceedings of the American Society of Civil Engineers*, April 1925.

American Committee, there are others which allow the deformations of the track and the vertical and lateral pressures exerted by the wheels on the rails on the straight and on curves to be measured.

This may be done, for example, by means of the very simple apparatus devised by the Belgian Engineers Flamache and Huberti, consisting of a lever, the end of the short arm of which rests under the head of the rail and the end of a long arm carries a pencil, by which one can measure the deflection of the rail joints under the effect of passing rolling loads. However, this apparatus is not very accurate, because the fulcrum of the lever is carried on a support resting on the ballast. More accurate results may be obtained by optical and photographic methods.

If it be desired to determine the maximum variation in the load on the springs, a pencil may be fixed to the axlebox and a sheet of paper to the horn blocks and the maximum displacement marked by the pencil measured. By using formula (30) one can calculate the maximum variation of the pressure of the wheels and axle, and consequently deduce the minimum pressure of the wheels on the rails, which is of interest from the point of view of security against derailment.

It is also very important, as regards the upkeep of the track, to determine the increase of pressure which takes place at the rail joints. This increase can be ascertained, either by fitting the axle under consideration with a maximum accelerometer of the Boyer-Guillon and Auclair type (1), or by placing under the sleepers close to the joints a dynamometer on the Brinell principle, which allows one to calculate, from the impression left by the ball in a soft metal, the pressure exerted by the wheels on the sleepers (2).

The maximum lateral forces on the track on the straight due to nosing action may be measured, either by Boyer Guillon and Auclair accelerometers applied to the axle, or by means of the Brinell type apparatus arranged between the axlebox slides suitably modified and the horn blocks.

The maximum lateral pressures exerted by the wheels on the rails at the entrance and exit of curves, in the case of an engine fitted with a bogie or truck which can be laterally displaced, may be calculated by measuring the maximum displacements of the bogie or truck. If the engine is not fitted with lateral control apparatus, one can determine these pressures by placing a Brinell type dynamometer between the rail and a suitably designed chair, or in any case by means of Boyer-Guillon and Auclair accelerometers applied to the axle.

In this way we may introduce into the formulæ which give us the coefficient of safety against derailment, the actual deflection of the track.

To obtain an idea of the easy riding of the vehicle, it is necessary to measure the maximum vertical and lateral accelerations, using, for example, the Boyer-Guillon and Auclair accelerometers. In order to avoid making a large number of tests, it is an advantage to employ a number of accelerometers, setting the spring of each of these to a different tension. It is also advisable that these instruments should be provided with an arrangement whereby they may be put into or out of action as required during the tests.

When we wish to investigate from a qualitative point of view the relative movements of the different parts of a locomotive or a vehicle, we may use the Sabouret apparatus with Marey pneumatic transmission, to transmit the movements which have to be investigated to the recording apparatus. The Italian railways use this apparatus for examining the track.

By following Mr. Marié's line of investigation, and taking as a guide the various

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(1) See *Comptes rendus de la Société des Ingénieurs Civils*, July 1913.

(2) See Marié : *Traité de stabilité du matériel des chemins de fer*, p. 479 and onwards.

investigations which have been carried out up to the present on transition curves, we can construct both track and rolling stock which are suitable for high speeds. The experiments which we have mentioned allow us to determine whether we are successful in obtaining the stability and easy running which is our object. This in our opinion is a method to apply in order that we may gradually obtain on the railways the increases in speed which have been so much under discussion, especially recently.

If we examine Mr. Marié's formulæ, which we have given in the first part of this investigation, we see that all the expressions connected with the oscillations due to deflections of the track are independent of the speed. It is true that the variations in the pressure of the wheels on the rails increase in the neighbourhood of the critical speed, but the effect of these deformations has been calculated for the critical speed, and we know that it decreases when once the critical speed is exceeded.

Further, the formula which gives us the lateral oscillations produced by the play between the flange and the rail is independent of the speed between the limits for which the coefficient K has been experimentally determined, which may, as a maximum, become equal to unity. Here again, the expression which gives us a condition of derailment is independent of the speed.

On the other hand, the formulæ which deal with the running of vehicles on curves and the oscillations of vehicles at the entrance or exit of a curve are all dependent upon the speed. However, these also may be considered as being independent of the speed, when, the allowable diminution in pressure on the inner rail being fixed, the permissible speed is increased directly as the square root of the radii of the curves.

The permanent lifting effect on the front end of the locomotive, due to steam action, increases slightly with the speed, because the required tractive effort increases.

The dynamic disturbance caused by the inclination of the outside rail above the inside rail along a transition curve is proportional to the speed, but, as we have seen, may be reduced by chosing suitable values for this inclination.

On the other hand, disturbances due to the inertia of the axles and, in general, of the non-spring-borne masses, and due to the gyroscopic effects of the rotating masses, increase proportionally to the square of the speed. Moreover, the coupling and connecting rods may fail by bending when excessive speeds are attained.

We have already said that for technical and economic reasons it is not advisable to exceed, with steam locomotives, the speed of 130 km. (80 miles) per hour. With multiple unit electric traction, we can eliminate a number of the causes which in the case of locomotives, prevent this speed being exceeded.

By transmitting the power by means of gearing and a quill drive, it is possible to do away with rods and to employ driving wheels of small diameter, reducing considerably the inertia effects on the track of the non-spring-borne masses. With this system of transmission and with spring-borne motors, we can not only give the motor bogies an elastic lateral displacement with frictional devices for damping out the lateral shocks on the track as far as the spring-borne masses are concerned, but by providing the axleboxes with inclined planes, we can allow the axles a lateral displacement relative to the frames which serves to reduce the lateral shocks on the track due to the mass of the bogies and the motors. Provided that the lateral shocks on the track can be satisfactorily damped out, even in the case of synchronism of the nosing oscillations, and that the use of springs is no longer necessary to reduce the lateral pressure, we can lower the centre of gravity and thereby improve the stability of the vehicle against overturning. Moreover, the axles being situated at the extremities of the frame,

the lateral reactions of the track act with a larg· leverage, which reduces the actual value of the reaction.

Mr. Marié has calculated the stability of a 60 t. multiple unit electric bogie coach, both the axles of each bogie being driving axles, the drive being by gearing and a quill, and with a lateral displacement of the bogies and of the axles. He has found satisfactory coefficients against derailment and overturning for speeds as high as 200 km. (125 miles) per hour, assuming curves of 1 500 m. (75 chains) radius, a maximum superelevation of the outside rail of 150 mm. (5 29/32 inches) on a curve, transition curves 100 to 150 m. (5 to 7 1/2 chains) long, a play between the flanges and the rails of 20 mm. (25/32 inch), and the deflection at the rail joints under the effect of passing loads not exceeding 6 mm. (15/64 inch).

Assuming that lines were constructed with these characteristics, and in consequence at great cost, trains could run over them at very high speeds, and by adopting a larger amount of superelevation, the speed could be further increased or the radius of the curves reduced.

We know that during the tests carried out on the short line from Marienfeld to Zossen, single electric coaches weighing about 100 t attained a speed of 200 km. (125 miles) per hour,

To conclude, theoretical and practical investigations can be carried out for lines which have to be run over at high speeds and also for the rolling stock intended for such lines. It will be prudent to proceed slowly introducing into the permanent way and rolling stock the modifications found advisable by experience, so as to enable the staff to become acquainted therewith and take the necessary precautions to ensure regularity of working.

[621 133.7]

The past, present and future of water treatment.

(*Railway Age.*)

The treatment of railroad water supplies formed the subject for consideration by one of the sessions of the annual convention of the American Water Works Association at Buffalo, N. Y., on 9 June 1926. Under the auspices of the Committee on boiler feed water studies, three papers were presented on phases of this subject by S. C. Johnson, chief chemist, water supply department, Chesapeake & Ohio; R. E. Coughlan, supervisor of water supply, Chicago & North Western; and C. R. Knowles, superintendent of water service, Illinois Central System.

These papers are abstracted below.

A review of the developments to date,

By S. C. JOHNSON,
CHIEF CHEMIST, WATER SUPPLY DEPARTMENT, CHESAPEAKE & OHIO.

The quality of the water supply is closely associated with the operating efficiency of a railroad and its result has a marked influence on the locomotive maintenance expense. Of the 350 billion gallons of water now being

used annually for steam purposes on American railroads, it is estimated that 50 billion, or about 15 %, is receiving treatment in some form. At a general average cost of 4 cents per 1 000 gallons for treatment, the yearly operation expense is in the neighborhood of $2 000 000. There are approximately 1 200 water stations, out of a total of 16 000, where chemicals are added and the total investment in softening plants, including the inexpensive as well as the elaborate types, is at least $10 000 000. It is estimated that these plants are removing 100 000 000 lb. of scale-forming solids annually, which, if allowed to enter the locomotive boilers, would involve an additional expense in locomotive operation and maintenance of approximately $13 000 000.

The properties in water which cause trouble in stationary power plants are of similar concern in locomotive practice, although the refinements are not, as yet, applicable. The chief concern seems to be the non-carbonate scale which causes damage by forming a hard cement-like coating on the heating surfaces with consequent deterioration of the material and loss of fuel. The carbonates of calcium and magnesium are troublesome when present in large quantity, due either to increasing the bulk of the scale or precipitate. Waters are occasionally encountered which initiate pitting and corrosion troubles if not corrected. In some parts of the country, particularly that section commonly known as the alkali plains, it is necessary to use waters high in alkali salts, principally sodium sulphate, which not only aggravates the foaming tendency of the waters, but also contributes to an electrolytic condition which accentuates pitting and corrosion unless counteracted. It is possible that silicious matter has entered into the scale problems in some quarters but the complaints from this possibility have been of minor importance and it is only within a recent period that serious consideration has been given to the elimination of mud and suspended matter.

Early development most rapid in Middle West.

The development of water treatment on American railroads has received its greatest attention on middle western systems where the objectionable quality of the natural waters was such that some form of treatment early became an operating necessity. Efforts toward correction were confined at first to treatment, with so-called boiler compounds or metal treatments with secret formulæ.

The next development in counteracting the harmful impurities was in the application of soda ash, both direct in the boiler or at wayside tanks. The scale reduction properties of this material are well known and this appears to be the least expensive systematic treatment from a first-cost standpoint. However, the precipitate formed by this reaction is so finely divided that the sludge in the boilers, together with the increase in alkali salt concentration, causes serious foaming conditions and the system has been found objectionable unless followed up with careful supervision to insure the boilers being blown down sufficiently to maintain the concentration within workable limits.

The systematic treatment with soda ash at wayside tanks appears first to have been developed on the Chicago, Burlington & Quincy and later extended to the Wabash, the Frisco, and the Alton, while several other roads are using it to some extent. The method consists in treating all waters, where necessary, to insure an excess of about two grains per gallon of sodium carbonate. A frequent check is made of samples from locomotive boilers and an effort is made to maintain approximately 15 %

of the total dissolved solids as sodium carbonate. This internal inspection is also necessary to insure sufficiently frequent blowing to keep the total dissolved solids below 125 grains per gallon in order to prevent foaming delays. Where sufficient supervision is provided to insure the carrying out of the predetermined practice, results have been secured. However, the necessity for the careful check and follow-up of the locomotive operation presents such difficulties that the extension of this system has been somewhat limited.

Experiments with zeolite softening for railway service have not been altogether satisfactory although tests being made at the present time on the western coast indicate possible success with certain types of waters. The preponderance of surface supplies with their occasional high content of suspended matter, limit the scope of this system as well as the dissolved solids quality, unless prefiltration is provided, with additional expense for installation and maintenance.

Methods.

The better established method of railway water softening consists of the addition of lime and soda ash to the water in predetermined amounts at wayside settling tanks. Its object is not only to soften the water but also to remove the precipitated sludge with mud or suspended matter, so as to deliver the waters to the boilers not only soft but clean.

The lime and soda process of treatment in railway service has developed from the simple intermittent system which consisted merely of two or more tanks which were filled, treated and used alternately, through the intricate proportioning devices with continuous automatic proportioning, back to the more simple continuous systems.

The chief essentials are proper chemical proportioning and sufficient mixing and reaction time followed by sedimentation and clarification before delivery of water for use.

Many intermittent systems are still in use and the capacity of treating tanks varies from 10 000 gallons to 500 000 gallons, but the governing principle is the same and similar satisfactory results can be secured. The usual means of agitation is by compressed air.

Plants of the continuous type consist of large tanks, usually of steel, with inside tubes of sufficient size to retain the water during the mixing and reaction period of from 30 to 45 minutes. The water and chemicals are mixed in these tubes in continuous proportion, flowing from the mixing tube to the bottom of the sedimentation tank from which they rise to a predetermined point before the clear water is drawn off for service. The specific gravity of the precipitated sludge is sufficiently greater than water to permit clarification in five hours if the vertical velocity of the settling water does not exceed 8 feet per hour, provided, of course, that the proper amounts of chemicals have been added to insure complete reaction and unbalanced equilibrium is avoided. If clarification troubles are experienced, filters are sometimes provided. Some experimenting is again being done with the elimination of the downtake tube, merely running the chemicals and water together without special mixing or agitation at the bottom of the sedimentation tank, although experience with this system some years ago was not entirely satisfactory.

In any system of water softening for railroads, the largest single factor in securing satisfactory results lies in competent and interested supervision. The chemical quality of the raw and softened water should not only be checked at frequent intervals but inspection should also be made of the mechanical facilities

to insure dependable and uninterrupted service. On railroads where treatment is practiced to any appreciable extent, systematic methods are therefore necessary to permit the plants to be operated with a minimum force.

Savings resulting from water treatment.

The savings possible through improvement in the quality of railroad water supplies are necessarily dependent upon local conditions. In 1914, the American Railway Engineering Association presented figures to show that the cost of each pound of incrusting matter permitted to enter the locomotive boiler, in such condition that it would deposit as scale on the tubes and sheets, was 7 cents, considering only the effect on fuel consumption, boiler and roundhouse repairs and engine time. This figure, transposed to present day prices, is 13 cents. Study by a special committee of the American Railway Engineering Association for the past 5 years indicates that this figure is conservative. There is no question but that, with proper treatment of the water, scale and pitting conditions, with their incident boiler maintenance expense, can be largely eliminated and that the fuel consumption in clean and dry boilers is much less than with leaky or badly scaled power. In addition, the large intangible benefits, such as the elimination of engine failures on the road and the reduction in delays to traffic and train movements, usually far outweigh the tangible savings in fuel consumption and boiler repairs. However, each individual case is a problem in itself and requires special study in order to obtain the best results.

On the Chesapeake & Ohio water softening plants were in service at 30 of the 207 water stations during 1925. Of these plants, 23 were of the continuous type, 2 intermittent and 5 of the simple soda ash system. Of the 6 135 922 000 gallons of water used for steam purposes, 2 496 038 000 gallons, or 40.7 %, was treated and a total of 2 672 080 lb. of injurious scaling and corrosive matter removed before the water was delivered for steam boiler use. The total cost for chemical treatment, including operation, maintenance, interest and depreciation, was $103 715 or an average of 4.5 cents per 1 000 gallons, while the cost for chemicals alone only averaged 1.98 cents. The estimated net saving for the year by reason of this treatment amounted to $243 675 which represents 67.2 % on the $356 323 invested in water treating facilities. This estimated saving averages only $480 per locomotive using the treated water which is a conservative rating.

The wide range of the problem of water treatment, including the design and installation of plants, as well as the individual studies of the water quality and the check of the actual treatment and its effect upon transportation and train movement, warrants the special study which is being given to it on many roads. In the handling of this problem over the wide range of territory involved on even the smaller railway systems, it is necessary to have the closest co-operation between the chemical supervision and the men actually operating and maintaining the plants, as well as the interest and assistance of the motive power and transportation departments, to secure the best, or even passable results. The best mechanical facilities will function only in a perfunctory manner or fail entirely if not followed up by a careful check system, and it has been found that an organization with a definitely fixed responsibility is the first essential to successful results from a railway water treatment.

A Statement of the problems now confronting the water service engineer,

By R. E. COUGHLAN,

SUPERVISOR OF WATER SUPPLY, CHICAGO & NORTH WESTERN.

On the Chicago & North Western practically all of the water supplied to boilers, except that obtained from lakes and rivers in northern Wisconsin and northern Michigan, requires treatment. Partial treatment and internal treatment have been used for many years. The extensive water treatment program of this railway was started in 1903 when 16 lime and soda ash treating plants were built in Iowa, where the water is hard, due to the large quantity of sulphates of magnesia and lime contained in solution. These were the pioneer water softening plants in that section of the country and have been added to from time to time, until at present 47 lime and soda ash water softeners are in operation and 10 more are under construction. These are supplemented by partial treatment and internal treatment where local conditions warrant. The intermittent type of softeners was the first installed. In 1922, the first continuous type softener was installed. Where internal treatment is used, this treatment is controlled by means of chemical tests made on samples of water taken from the boilers at each terminal. Simple feeding devices regulate the amount of treatment.

The boiler failure report of the Chicago & North Western for 1910 shows 2 132 failures, chargeable to water conditions. The same report for 1925 shows 37 failures. The monthly boiler failure report for February 1910, shows as follows :

Foiler failure report for month of February 1910.

Cause of failure.	
Leaking flues.	319
Leaking fireboxes	22
Leaking arch tubes.	0
Flues burst	3
Arch tubes burst	30
Foaming	17
Total. . .	391

The same report for February 1926, shows as follows :

Boiler failure report for month of February 1926.

Cause of failure.	
Leaking flues.	0
Leaking fireboxes	0
Leaking arch tubes.	0
Flues burst	0
Arch tubes burst	0
Foaming	0
Total. . .	0

Water treatment is one of the essential factors which make it possible to obtain the longer daily mileage which most railroads are now receiving from locomotives in passenger service. A few years ago, it was the practice to change locomotives about every 100 to 150 miles. Now many railroads that are equipped with water softening plants use one locomotive for continuous runs of much longer mileage. The Chicago & North Western now operates locomotives in passenger service from Clinton, Iowa, to Omaha, Nebraska, a distance of 350 miles, which make a round trip of 700 miles each day. Another western railroad operates locomotives in passenger service 600 miles without change, while still another has completed test runs of over 1 700 miles, one locomotive pulling the train the entire distance.

Progress in reduction of corrosion and pitting.

With the problem of incrustation of boilers practically under control, more attention is now being given to the pitting of flues and the corrosion of boiler sheets. Many theories as to the cause of this trouble have confused the issue. Many committees have held symposiums on the subject. The railroad water service engineers have not been idle, although their work has been somewhat hampered due to incomplete records and the increasing evaporating power of boilers. When it was found that removing the scale did not prevent corrosion, further studies have been made along three special lines, namely: 1) Use of feed water heaters to eliminate oxygen; 2) counter-electrical potential devices, and 3) excess treatment.

The first two of these methods are still in the experimental stage, each method having its respective merits and adherents. The excess treatment method has shown great possibilities where it is practicable to apply it in railroad service. This method, proven in the laboratory, consists in the addition of an excess of caustic soda, caustic lime or sodium carbonate over that required to combine with the scale-forming salts. The success of this treatment depends upon uniformity of treatment over an entire locomotive district to prevent foaming. Where it is possible to secure this uniformity, high concentrations of alkaline salts are carried in the boilers with practically no foam trouble.

The Chicago & North Western has a locomotive district where 50 % of the natural water contains over 50 grains per gallon of sodium carbonate. By treatment of the remaining water to a similar composition, a concentration of over 8 % normal alkalinity is carried without trouble. Pitting and corrosion on this district are unknown and needless to say the boilers are clean. Additional lime and soda ash softeners are being installed on other locomotive districts as rapidly as funds become available, so that in a short time this method of treatment will be in general use.

One of the railroads having 71 lime and soda ash softeners in operation is treating the water to less than one grain per gallon hardness, leaving in the water an excess of lime and soda ash for the purpose of eliminating pitted flues. This program was started early in 1922 and very gratifying results have been obtained. The boilers carry a very high percentage of alkaline concentration with very little trouble.

The quality of the material used in boiler construction has also been thoroughly investigated. At the present time, most of the railroads have standard specifications for boiler steel. These specifications are strictly adhered to in order to avoid the use of non-homogeneous steel which may set up electrolytic reaction, leading to corrosion. Protective coatings of lead or similar material have also been used with varying degrees of success.

Progress through co-operation.

While the entire principles of water softening and the control of corrosion may be explained theoretically in very simple terms, practicable applications are sometimes very difficult. Experimenting with the boiler of a locomotive in operation is entirely different from research work conducted in a laboratory. Foam trouble and priming cannot be tolerated. The water service engineer must be sure of the results. The movement of trains safely is the first consideration. Progress can only be attained with the co-operation of all departments.

A forecast of the probable future development of railway water treatment,

By C. R. KNOWLES,
SUPERINTENDENT WATER SERVICE, ILLINOIS CENTRAL SYSTEM

The railroads have passed through the period of pioneer development in the treatment of locomotive water supplies and are now approaching an era of refinement that will undoubtedly lead to still greater development. The foundation of water softening and treating practice has been laid and it is unlikely that there will be any radical departure from these principles. At the same time it is becoming more and more apparent that refinement must take place in many of the present methods of treatment and that careful consideration should be given to the possibilities of new methods.

As far as the railroads are concerned, water treatment has not only been an economic necessity but in many cases it has been an actual operating necessity. As train loads became heavier and boiler pressures and ratings were increased the treatment of many of the worst waters became imperative. The available information regarding methods of treatment was limited, and many of those who first undertook to treat the water were confronted with the problem for the first time and were probably in the same state of mind as the first man who started out to soften water more than eighty years ago, the idea in the beginning being simply to get the water in a tank, add the necessary chemicals and let it settle. Certain refinements were developed through experience as time went on and it may be expected that the art will continue to develop by the same process.

It was quite natural that the worst waters should receive attention first for while the less troublesome waters of comparatively low scaling content were costly from the standpoint of fuel economy and boiler maintenance, they did not interfere seriously with train operation. As treatment has been extended to include practically all of these very bad waters, the water that was considered fair a few years ago has become the bad water of today.

The value of water treatment has become so well established through records of savings in boiler repairs, fuel economies and the reduction of engine failures that it is becoming apparent that treatment must be extended to include these waters of so-called fair quality. The cost of providing adequate motive power for the movement of trains is constantly increasing, and with investments of from $60 000 to $70 000 each for locomotives, poor water conditions can no longer be tolerated.

Longer engine runs are another factor in locomotive operation that is making better water conditions necessary. The Northern Pacific recently made a record-breaking continuous freight run in which a Mikado type coal-burning locomotive pulled a train from Seattle, Washington, to St. Paul, Minnoseta, without being uncoupled from the train and without receiving any terminal attention en route. The distance of 1 898 miles over three mountain ranges with maximum grades of 2.2 % was covered in 109 1/2 hours total time. The Great Northern ran an engine with a fast mail train from Seattle to St. Paul and return, a total distance of 3 462 miles, a short time ago, while the Burlington handled a train of 90 cars loaded with 6 450 tons of coal from West Frankfort to Chicago, a distance of 413 miles with one Mountain type locomotive. The Southern Pacific is running passenger trains regularly between El Paso and

Houston, Texas, a distance of 832 miles, without changing engines. While many of these long runs are test runs today, they will be a regular thing in the near future.

All of these developments in locomotive operation call for water supplies of the highest quality, and it is evident that the railroads are going to demand much more of the water service engineer in the future than they have in the past. Some of the improvements in the quality of water will no doubt be accomplished through the development of new and better supplies, but in the great majority of cases it will mean the improvement of existing facilities by establishing additional treating plants, etc.

The removal of suspended matter.

One of the problems that will undoubtedly receive more attention in the future than it has in the past is the removal of mud or other suspended matter from railroad water supplies. While mud removal has not been altogether neglected, it has hardly received the attention it deserves, to say the least. This apparent neglect is not due entirely to failure to appreciate the importance of removing suspended matter, but rather to the fact that through necessity efforts were concentrated on improving waters high in dissolved solids, and in some instances to lack of sufficient finances to provide the required facilities.

The rivers of the middle west carry an average of from 1 to 1 1/2 pounds of suspended matter per 1 000 gallons of water throughout the year with a maximum of 5 or 6 pounds during the freshet seasons, while the southern and some of the western streams carry still more. The objection to the presence of mud or other suspended matter in boiler waters is not altogether due to the necessity for frequent washing of boilers to prevent foaming and possible mud burning of sheets due to the use of turbid water, but also to the fact that the mud is deposited on the tubes and sheets and in the presence of dissolved solids helps to form scale.

Mud removal installations consist largely of sedimentation basins alone. They are fairly effective where the mud is heavy and settles rapidly but the most satisfactory results can be secured by coagulation and sedimentation, followed by filtration. Most turbid waters carry considerable amounts of scale-forming solids in addition to the suspended matter and as softening the water as well as removing the mud add little to the cost, the development along this line will probably include complete softening equipment for the removal of matter in solution as well as in suspension.

Filters.

Another possible future development of water treatment is the more extensive use of filters. A large majority of the railway water treating plants depend entirely upon sedimentation to produce a clear treated water, although the limitation of sedimentation alone in clarifying the water is recognized in the extensive use of iron and alum in various forms as an aid to the precipitation of the sludge.

It is a commonly accepted fact that the temporary hardness remaining in water after being softened with lime and soda ash is materially lowered by filtration through sand. Where filters are used in conjunction with lime and soda ash softeners no other coagulant is required, as a general rule. The amount of suspended solids that may be removed by filtration, even after treatment and sedimentation, is of appreciable value, not only in reducing the hardness and preventing after precipitation in distribution systems, but also in reducing the losses in fuel and water through less frequent blowdowns of boilers. Filtration may well be compared to a

continuous blowdown in its effect, as the suspended matter that would otherwise remain in the water after ordinary treatment is being removed constantly as the water is passing through the filter. The additional cost of filters has been fully justified by the roads that are using them and their universal adoption as an adjunct to water softening will undoubtedly follow as their value becomes better recognised.

Treating tanks.

Improvements in the design of water treating plants will no doubt include a more efficient type of treating tank, particularly in regard to sludge removal. Many different designs have been followed in the construction of reaction and sedimentation tanks, the most widely used being the common flat bottom tank, either of the standpipe type with the bottom at the ground level, or erected on an elevated tower. In this type of tank, as commonly used, the sludge is deposited over the entire bottom of the tank, its removal necessitating the construction of an expensive sludge-collecting system, which at its best leaves much to be desired. This sludge removal system usually consists of an arrangement of pipes with openings placed at uniform intervals over the bottom of the tank. In the larger tanks they are so arranged that they may be operated in sections with from 2 to 4 sludge valves for each tank. Another type is so arranged that the collecting pipes may be rotated over the bottom of the tank. None of these systems is entirely satisfactory for they do not completely remove the sludge and they are extremely wasteful of water, as the sludge nearest the outlet is the first to be picked up and two-thirds of the openings are usually discharging clear water before the sludge at the outer edge of the tank is disturbed.

One of the principal developments in the design of treating tanks is the conical bottom tank with a large riser pipe. The angle or slope of the cone is 45°, and the drum or riser pipe is from 4 to 6 feet in diameter and serves as a sludge-collecting chamber. The sludge falls to the bottom and is collected in the mud drum as it forms, where it is easily and completely removed through a single opening and with a minimum waste of water. Actual tests to determine the amount of water used in removing sludge show that the flat bottom tank requires from 5 to 10 times more water in sludging than a conical bottom tank of proper design.

The remarkable efficiency of the conical bottom tank in sludge removal and economy in the use of wash water are factors that will result in its more extensive use for water treatment. Experiments are also being conducted with continuous sludge removal systems and tanks of other designs, and it is evident that the development of water softening will include tanks of more efficient design.

Zeolite softeners.

Zeolite water softeners are receiving considerable attention as a possible method of treating water for locomotives. One installation on a western road is being watched with much interest and the results obtained will undoubtedly be a factor in determining their use by other roads, as well as extending their use on the road in question.

The chief concern regarding the use of zeolite is on account of the possibility of increased foaming due to the conversion of the various elements in the untreated water to sodium compounds in the treated water which is characteristic of zeolite treatment. It is claimed that the foaming may be minimized with the exclusive use of zeolite treated water, which is practically free from suspended matter. This is not always

possible on a railroad unless entire engine districts are equipped with zeolite treatment. Another factor which has reacted against their use for railroad supplies is the fact that with certain waters pre-treatment is necessary. For example, if the water contains an appreciable quality of iron it must be pretreated before passing through the zeolite softener, or if the water is turbid it must be filtered before it can be handled successfully. With the lime and soda ash plant no pre-filtration is required, as sand filtration after reaction and sedimentation is provided in all modern equipment.

A possible solution of the problem is a combination lime-soda-zeolite plant with pre-treatment by lime and soda ash for the removal of the bicarbonate of lime and magnesia, iron and free carbon dioxide, the neutralization of sulphates and the reduction of organic matter, silica and alumina, followed by the zeolite treatment to produce a water of zero hardness with a minimum of suspended matter. If this type of plant can be developed successfully and economically it is not at all unlikely that zeolite treatment will be further extended in the treatment of locomotive supplies.

Internal treatment.

Internal treatment of boiler waters involves the introduction of chemicals and other substances into storage tanks, the tenders of locomotives or direct into the boiler. This method of treatment as applied direct to boilers is the oldest form of treatment known, probably being as old as boilers themselves. It has been improved upon from the old hit-and-miss form of treatment until it has become a generally accepted process. Reputable manufacturers offer forms of treatment that are based upon chemical examination of the water and that are producing good results. The combination of chemicals used in this form of treatment is generally referred to as boiler compound. The future development of this form of treatment will be the elimination of the quack cure-all compounds and more uniform methods of feeding the chemicals to the water, probably through a more extensive use of compound feeders.

General.

Stationary power plants are adapted to the use of many devices for the purification of boiler feed waters that cannot be used on locomotives on account of the limited space available. Therefore, it is impossible to make any prediction as to the probable future use of de-areators, de-concentrators, etc., on locomotives, although their development for this purpose is not beyond the realm of possibility. In fact many of the feed-water heaters used on locomotives embody the feature of oxygen removal in addition to heating the feed water. Some authorities claim great benefits from this feature in preventing corrosion, and it is possible that as more locomotives are equipped with the open type of feed water heater they will become an important factor in improving the quality of the feed water. The claim is also made that thermic syphons and plates as installed in locomotive boilers tend to improve feed water conditions, particularly as regards corrosion, through increased circulation of the water in the boiler.

It is impossible to make any prediction as to the possible future development in the use of the less common reagents in water softening, such as the barium salts, for example. As with many other reagents, treatment with barium is not yet commercially practicable as a rule although barium salts may ultimately be manufactured at a cost that will result in their more general use. However the principal developments, in the near

future at any rate, will be based upon the lime soda ash treatment.

There is ample room for future development in railway water treatment; at the same time there are certain limitations that must be considered. Water softening and treatment for steam making must of necessity be kept within the bounds of the more simple reaction and commercial reagents.

[656. 253 (.75)]

New York Central installs Miller train stop.

Figs. 1 to 17, pp. 158 to 163.

(*Railway Signaling.*)

The New York Central Railroad has equipped 29 miles of track and 10 locomotives with the alternating current intermittent induction train stop of the Miller Train Control Corporation. Between Air Line Junction interlocking and the end of the New York Central northbound track at Alexis, Mich., a distance of 7.5 miles, and between Monroe, Mich., and Air Line Junction interlocking, a distance of 21.5 miles on the southbound track, a total of 34 inductors were installed at automatic and interlocking signals. Portions of this equipment have been under test operation for several months and the complete installation was placed in full service on 1 April.

The Miller intermittent inductive train stop system, as being used on this New York Central installation, incorporates several features of the Miller ramp system as used on a division of the Chicago & Eastern Illinois for 11 years, and which has been approved by the Interstate Commerce Commission. For example, the apparatus for actuating the engineman's brake valve and the procedure of forestalling are the same as those on the Chicago & Eastern Illinois. The pneumatic control feature of this actuator, which in the ramp system is controlled by the shoe, is, in the case of the inductive system, controlled by an electro-pneumatic valve, the electrical features of which are governed by the inductive apparatus, as will be described later.

The track element is so mounted as not to extend above the level of the rail, and, therefore, it does not interfere with the operation of snow plows or spreaders, nor is it liable to be struck by open dumpcar doors or low parts of other equipment. The receiver operates with an air gap of 4 inches, therefore it is mounted on the locomotive 7 inches from the gage of the rail, and as the receiver is 10 inches wide this brings the entire locomotive apparatus within 17 inches of the gage of the rail at a height of 4 inches. Thus it may be seen that this inductive system has the advantage that the track element may be placed away from the rail , so as to be out of the way, and the receiver can be mounted near the locomotive and placed high so as not to interfere with misplaced maintenance of way material along the track. Such mountings, of course, bring the track element and receiver off center 6 inches, however this difference in alignment is no obstacle because of the operating principle of the alternating current magnetic balanced receiver, which also permits the large air gap of 4 inches.

The simplicity of the system is shown

by the fact that it employs only three major pieces of engine equipment, *i. e.*, the receiver, the electro-pneumatic valve and the engineman's valve actuator. The electro-pneumatic valve is dissimilar from an ordinary relay in that the electro-magnetic field is so strong as to operate and hold its armature against the valve which is under air pressure at 130 lb. This armature has a solid brass bushing, forced in a hole at the center. The bushed armature is free to revolve 3/4 of an inch on a 1/8-inch Stubb steel shaft which is fixed in the legs of the frame.

An outstanding characteristic claimed for the Miller inductive system of train stop is that the principle of operation, the circuits and conduit layouts are so designed that any cross or ground on any of the wires between parts of the apparatus will not cause a false clear operation. An equally important characteristic claimed for this system is its immunity to interference from stray currents, third rails, reactance bonds, frogs, bridge members, water troughs, tie plates, switch plates, or solid bodies along the track.

Description of apparatus.

The Miller type-HB induction train control as installed on the New York Central is of the intermittent alternating current inductive type with normally inert track elements. No battery or other local energy is required for the track elements. These latter are controlled by the signal control relays and so far as possible they repeat the indications of the signals.

An « H » shaped magnetic circuit with two parallel iron cores and two primary and two secondary windings is employed in the receiver on the engine.

Heavy wrought iron brackets 1 inch by 5 inches support the receiver from the rear end of the right frame of the locomotive. These brackets are so designed as to extend down to the receiver at different angles, thus tending to eliminate periodic vibration.

In passing over a stop inductor the distribution of flux is changed in such a manner that the electro-pneumatic valve on the engine is de-energized. This is explained in principle by the diagrams in figure 4 and figure 5. These show two laminated bars arranged in the shape of the letter « X », one containing a primary winding, the other a secondary winding. The combination turbo-generator supplies alternating current at 32 volts and 360 cycles to the primary coil, while the secondary coil is connected to the electro-pneumatic valve. The magnetic flux travels, normally, across the air gaps between primary and secondary cores as shown, completing a figure « 8 » path. This transformer action induces sufficient current in the secondary coils to hold the magnet valve in the proceed position. It is necessary that the two bars be less than 90° in displacement in order that the proper flux will embrace the secondary circuit. The electro-pneumatic valve must be energized sufficiently to prevent the escape of air at the head of the valve.

Action at a stop inductor.

On passing over a stop inductor, which consists of a pair of parallel laminated iron cores there is a marked change in the flux distribution and the flux in the secondary core tends to reverse due to the directional effect of the two track inductor cores. An actual reversal of secondary flux (with respect to a certain direction in the primary core) would reduce the valve current to zero momentarily and this would permit the valve to open. In practice there is no appreciable rise in current after passing through zero value following this reversal of flux in the secondary core. As the distance between the receiver

Fig. 1. — Passenger engine equipped with Miller inductive train stop in Chicago station.

Fig. 2.
The inductor is level with the top of the rail.

Fig. 3. — View showing off-set of receiver and inductor and clearance from third rail in Detroit terminals.

and track inductor is increased the valve current will tend to reduce to zero or nearly so without any rise after passing through this value. The limiting height of engine receiver is that at which the valve current while not approaching zero is yet sufficiently low to cause the armature of the valve to open. Variations in air gap due to action of the locomotive springs will not interfere with the operation of the system.

In the type-HB receiver as installed on the New York Central the two cores are not crossed, but the magnetic flux traverses the same figure « 8 » path as in the diagram of figure 4. In doing so the magnetic flux travels through four air gaps instead of two which allows the cores to be placed closer together while still retaining enough reluctance in the magnetic circuit to permit of a decided change in flux distribution when the receiver is over track inductors. The path of the flux is shown in the lower part of figure 6. This type of receiver is more effective than the type shown in figure 5 and its width and weight are less.

Resonant circuits have been introduced by the inclusion of condensers in the engine primary and secondary circuits with a resultant improvement in power factor. It has been found that a resonant condition assists in the reduction of valve current toward zero value at stop inductors and further prevents any appreciable rise of valve current beyond zero if an inductor causes a relative reversal of flux in the secondary core. Any magnetic or electrical derangement of the receiver or other part of the engine equipment will detune the normally resonant circuits. As a further precaution the conduit system is so arranged that no crosses, short circuits, or grounds on the engine circuits can tend towards false clear conditions. The connections of both primary and secondary circuits on the engine are shown in figure 7.

At a clear inductor

Embedded in each track inductor are six coils which are controlled through the signal control relay and « reflecting» condensers. The track inductor is placed 30 feet from the signal. When a clear condition exists these coils in the inductor are connected to the condensers and upon the approach of an engine receiver the energy is « reflected » back to the receiver with little change in flux distribution. This action maintains the valve current as the receiver passes over the inductor.

At a stop inductor, as explained previously, the two inert parallel iron cores of the track element cause a decided change in the distribution of receiver flux. This action occurs automatically when the signal control system open-circuits the inductor coils.

Power supply system.

Energy for the train control is obtained from a Pyle-National type E-3-M turbo-generator which is rated at 800 watts direct current for the headlights and cab lights and 300 watts alternating current at 32 volts and 360 cycles for train control. The actual requirement of the train control system is about 90 volt-amperes.

Brake actuator.

In the type-HB induction train control the same arrangement is employed for operating the handle of the regular Westinghouse or New York engineman's air brake valve on the locomotive as is used on other Miller train control installations. The brake valve actuator is shown in figure 8, which also illustrates how it is governed by the Miller electro-pneumatic valve shown in the lower right hand corner of the drawing. Normally, the armature of this valve is held in the closed proceed position by the valve current flowing in the windings. This armature working through a

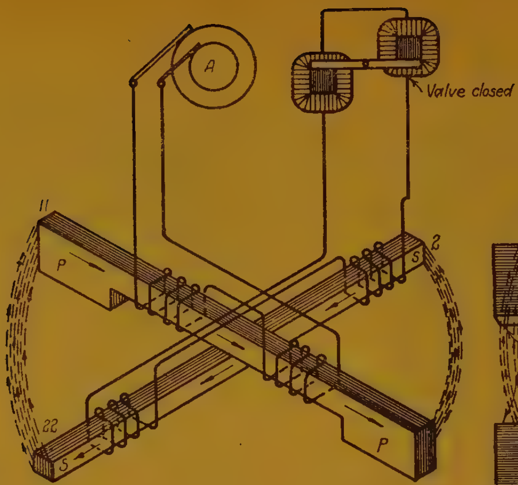


Fig. 4. — Normal receiver action.

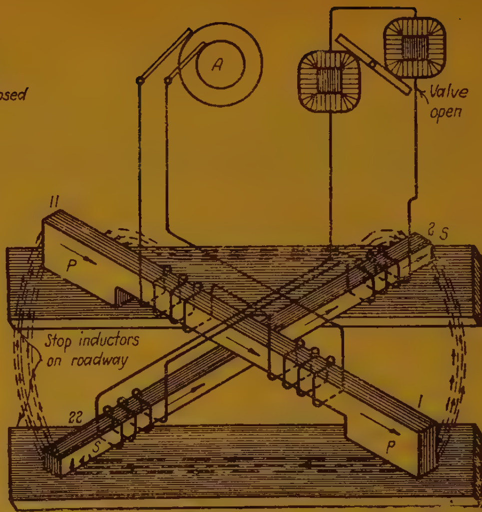


Fig. 5. — Acting on passing stop inductor.

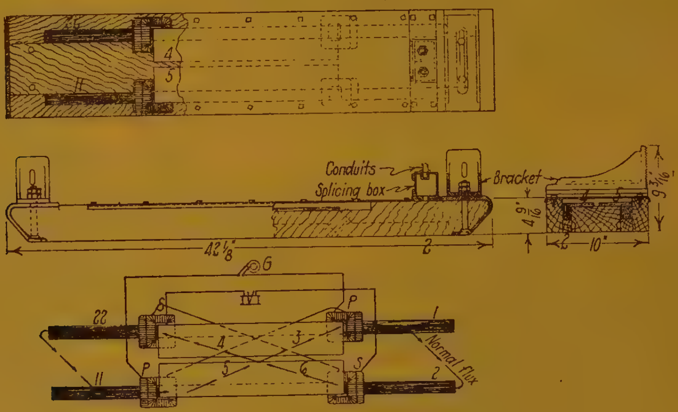


Fig. 6. — Receiver construction.

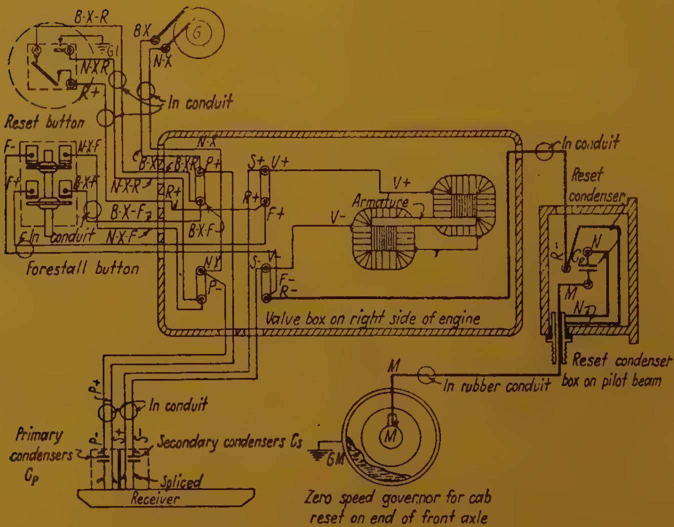


Fig. 7. — Engine circuits.

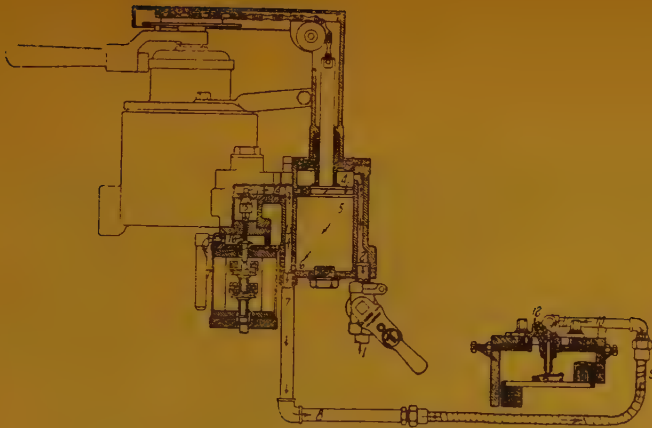


Fig. 8 — Pneumatic diagram of train stop equipment.

Fig. 9.
Actuator on engineman's valve.

Fig. 10. — Side view of inductor, receiver and electro-pneumatic valve.

Fig. 11. — Fluid governor mounted on end of front axle.

Fig. 12. — Electro-pneumatic valve with terminal board.

Fig. 13.
Inductor at interlocking signal.

Fig. 14.
Reflecting condenser in case.

Fig. 15.
Typical automatic signal location.

Fig. 16. — Reset button mounted in top of the cab above engineman's seat.

Fig. 17. — Special Pyle-National turbo-generator.

valve spring keeps the valve raised against the valve seat, preventing any discharge of air until a stop inductor is reached which, of course, will reduce the valve current and release the armature. To prevent the valve from leaking in case of wear in the valve seat, the valve spring was introduced as a slack adjuster. A spring suspension is used for the entire valve assembly, periodic oscillations being damped out effectively by leather strips fitted into the springs.

Forestalling and resetting.

Referring to figure 7, the engineman, to forestall an automatic brake application at a stop inductor, presses the forestalling button which connects the alternating current supply of the turbo-generator to the electro-pneumatic valve, thus retaining the latter in the proceed position regardless of the change in magnetic flux distribution occurring in the proximity of a stop inductor.

Once the valve armature is opened it cannot be restored to the closed position by either the secondary circuit of the receiver or the forestalling circuit, because the valve magnet has a high reactance. After an automatic application is initiated the train must stop, after which a reset may be accomplished. While the simplest method of doing this would have been to provide a mechanical push-button under the armature in such a location that the engineman would have to get out of the cab and on the ground to reset, it was felt that a reset operated from the cab was more desirable. This made necessary the inclusion of some positive motion detecting device to insure that the train had stopped before the cab reset could be operated. The Miller fluid governor is adapted to fill this requirement. It is mounted at the right hand end of the front axle and is so designed that any failure will give a more restrictive control, and furthermore any grounds and shorts in the reset circuit will prevent the engineman from resetting. When the train has been brought to a stop a reset can be effected by pressing the reset button, which connects the generator to the valve magnet through the reset condenser which neutralizes the reactance of the valve magnet.

MISCELLANEOUS INFORMATION

[621 .133.7 (.73)]

1. — Southern Pacific uses Zeolite softener, (1)

By C. W. STURDEVANT,

ASSISTANT ENGINEER OF TESTS, SOUTHERN PACIFIC COMPANY

(Railway Review.)

During the past twelve or fifteen years considerable attention has been paid to the Zeolite or base exchange method of water softening, especially for use in connection with laundries and stationary boilers. But practically no railroads except the Southern Pacific have made use of this method of treating water for use in locomotive boilers.

The principle of Zeolite treatment is that of base exchange. It makes use of base exchange silicates which possess those properties by which they may combine either with the base sodium or the bases calcium or magnesium. When regenerated, that is when ready to soften water, the Zeolite mineral is possessed with a supply of the base sodium in exchangeable form. During the softening run this base sodium is replaced by the calcium and magnesium bases of the hard water.

When the supply of base sodium is exhausted, that is, when the mineral has become possessed of its capacity of the calcium and magnesium bases this softening action ceases. At this stage of the process the mineral is ready for regeneration and is treated with a solution of sodium chloride (common salt). This salt is cheap and readily obtainable.

During the regeneration the exchange of bases is exactly the reverse of that which takes place while softening the water. At this time the sodium base of the salt is taken up by the mineral which gives up its capacity of the calcium and magnesium bases in exchange therefore. These hardening bases combine with the chloride of the salt solution which, during the regeneration or salt brine flushing, is changed from sodium chloride to calcium or magnesium chloride. These chlorides being soluble cause no sludge and may be flushed off through any convenient drain. With the rapid rate mineral now used by most of the maufacturers of Zeolite water softeners, the time required for reconditioning the Zeolite mineral has been reduced to about fifteen or twenty minutes.

The two processes, softening the water and regenerating the mineral with salt brine, are repeated alternately with practically no loss of the mineral itself during years of service.

In illustrating base exchange softening some of the principal advantages of this method over the process of chemically dosing hard water is brought out. With chemical dosage, desired results are possible only when an excess of the chemicals are put into the water to drive the reactions to completion.

With the base exchange method the reactions of regenerating the mineral silicate are driven to completion by an excess of salt solution. This treatment is entirely independent of the actual water softening, which does not take place while regeneration is in process; hence what excess of salt solution is required, simply is washed away to the drain with the chlorides of calcium and magnesium; the excess never is carried away in the treating of the softened water.

During the base exchange process of softening the water, the exchange of bases sodium for calcium and magnesium is in the exact

(1) A paper presented at the Western Society of Engineers, Chicago, 4 October 1926.

relation of the combined powers of these elements; therefore over-treatment is impossible.

At present the Southern Pacific has in operation nine plants of the Zeolite or base exchange type of water softeners, and five more are under construction. The plants so far installed and those under construction all are of the Wayne or Permutit manufacture. All the plants used by the Southern Pacific make use of the green (Zeolite) sand found in New Jersey, which is cleaned, graded, and stabilized by a process which prepares it for service.

The only cost attached to this method of water treatment, other than the interest and depreciation on the investment, together with what little attention is necessary, is the salt required to regenerate the mineral. This salt requirement is proportional directly to the incrustating content of the water it is desired to treat, and is almost exactly one-half a pound of salt a grain of hardness a gallon for every thousand gallons treated. For instance, a water containing fourteen grains of incrustating solids a gallon will require seven pounds of salt for every thousand gallons of water treated.

The amount of the Zeolite mineral required is proportional also to the amount of water it is desired to treat between periods of reconditioning, coupled with the incrustating content of the untreated water. The converting power of the natural Zeolite or green sand is about thirty-five grains to the pound of mineral, under most favorable conditions, to thirty grains, under ordinary operating conditions. Consequently, the design of the plant requires a certain amount of engineering in order that its size and capacity may fit in with local conditions.

The waters treated on the Southern Pacific lines are by analysis quite similar to those found in other parts of the United States, and vary, naturally, as waters elsewhere do. At present we are treating water varying from eleven grains of incrustating matter and 5.48 grains of soluble matter a gallon up to 33.94 grains of the incrustating matter and 8.69 grains of soluble matter a gallon. In one case, that of the El Centro, Cal., city supply, the raw water is from the Colorado river and carries 26.7 grains of incrustating matter in addition to 20.58 grains of soluble salts.

In every case the total incrustating matter is converted into soluble salts through the Zeolite process, producing a zero hardness water except for a minor content of silica, alumina and iron oxide. In ordinary cases these three items will run from one-half of one grain to a maximum of three grains a gallon. This content does no harm, however, for it does not form scale in the absence of a binder such as the calcium and magnesium carbonates and sulphates, which, as explained, have been converted to sodium carbonate and sulphate.

However, the soluble matter produced by the treatment, added to the soluble matter originally contained in the untreated water, builds up a total of combined soluble and unsoluble matter which makes, what is termed ordinarily, a light water. This condition always has been feared because of the belief that much foaming and priming might result. Furthermore, there has been a fear that corrosion or pitting might be encountered under such circumstances. Again as a result of this type of treatment some have entertained the belief that caustic embrittlement might result.

It is true in some cases, where our locomotive boilers are scaled badly, we do have a little trouble from foaming immediately after starting a Zeolite plant. This is due almost entirely to the fact that the softened water has a dissolving effect upon the old scale, producing a heavy sludge within the boiler, which, if not blown out regularly and often, will cause foaming. If a reasonable amount of care is exercised in handling the blowoff cock, together with proper washing of the boiler, very little, if any, trouble is encountered. After a short time most of the old scale disappears and the light water condition with it.

In many places where we have started Zeolite plants we have experienced no trouble

at all. Two points in particular, Calexico and El Centro in the Imperial Valley, are in a very bad water district. No light water conditions were experienced and, notwithstanding the heavy traffic at the time the plants were started, no anti-foaming compound was necessary.

We have paid particular attention to the matter of pitting and corrosion. As a test, we have suspended pieces of new boiler plate, boiler tubes, and steel wire in the boilers, as close to the fire sheet as possible. We have left them in position from six months to a year, observing and inspecting them, and at the end of the tests have found the material in its original condition. We appreciate these tests are not entirely conclusive, nor are they the same or as severe as if the test pieces were a part of the fire sheet or tube. However, we are mindful still of the possible after effects, and are continually on the alert. So far we have observed nothing to sustain the pitting and corrosion theory.

While we do not attempt to contradict the caustic embrittlement theory, we have had this matter under constant observation, and so far have found nothing to sustain it.

It is not the intention to claim the Zeolite treatment is a cure-all for all waters. There is no doubt, where water excessively high in temporary hardness (calcium and magnesium and magnesium carbonates), is encountered, this hardness first should be reduced or precipitated by means of lime treatment before passing the water through the Zeolite plant. Such action will reduce the total soluble or alkali content after treatment to a minimum. The Southern Pacific has not as yet found it necessary to resort to this pre-treatment, because we are not attempting to treat by the Zeolite method waters excessively high in temporary hardness.

It might be of interest to readers of the *Railway Review* to know just what does happen when water is treated by the Zeolite method. For this reason a table of comparative analyses of the Los Angeles and Oakland waters where we long since have had Zeolite softeners is given to show the conditions before and after treatment.

	Los Angeles city water untreated.	Los Angeles city water Zeolite treated.	Oakland city water untreated.	Oakland city water Zeolite treated.
Calcium carbonate	8.51	None	7.46	None
Magnesium carbonate	2.16	None	2.57	None
Calcium sulphate	1.11	None	0.41	None
Magnesium sulphate	1.57	None	2.45	None
Calcium chloride	None	None	None	None
Magnesium chloride	None	None	None	None
Silica, oxide of iron, and aluminum	1.87	1.63	1.28	1.11
Incrustating solids	15.22	1.63	14.17	1.11
Alkali carbonates	None	11.31	None	11 37
Alkali sulphates	4.55	7.99	0.23	3.44
Alkali chlorides	2.33	2.04	5.19	5.19
Alkali nitrates	0.58	None	None	None
Non-incrustating solids	7.46	21.34	5.42	20.00
Total solids	22.68	22.97	19.59	21.11

The figures represent grains per U S. gallon.

It will be noted, as mentioned previously, that practically all the incrustating matter has been converted either to sodium carbonate or sodium sulphate. In the case of Los Angeles where the untreated water contained 15.22 grains of incrustating matter and 7.46 grains of soluble matter there are only 1.63 grains of incrustating solids in the treated water, all of which are silica or the oxides of iron and aluminum. But we do have 21.34 grains of soluble matter in the treated water. For all this the total content of solids

is substantially the same in both waters, because nothing has been added to or taken away from the water.

Notwithstanding the fact that sodium chloride (common salt) has been used in reconditioning the Zeolite mineral, none of this chloride is introduced into the treated water as is borne out by the analyses. The analyses, if absolutely correct, should show exactly the same amount of sodium chloride after treatment as the water contained before it was treated. This is done in the case of the analyses of the Oakland water. The difference between the sodium chloride content and untreated water at Los Angeles is slight.

In conclusion, actual observations thus far show marked reduction in maintenance cost of our boilers which are operated in Zeolite treated water districts. Savings due to reduction in boiler washings in turn make possible a more continuous performance of the locomotive or boiler, and it is hoped we may be permitted to install several more of the Zeolite type softeners in our bad water district in the near future.

[625 .143.4]

2. — The « S » spring plate.

A new device for preventing the slacking of nuts on railway fish plates.

Figs. 1 to 4, p. 169.

(Les Chemins de fer et les Tramways.)

A number of devices have already been invented for keeping fish-plated joints tight by preventing the slacking back of the fish plate nuts. Amongst these the « Grower » washer, and spring plates are the best known, although they do not fully meet the case.

As they are flattened out when tightened up, these washers and plates, in practice, soon lose most, if not all, of their elasticity, and then act as plain washers without any locking effect.

It is advisable therefore to call attention to the recently patented « S spring plates » which are being tested on the lines of the chief French and foreign railways. The Paris-Lyons-Mediterranean Company intend to extend the trials which were started about a year ago. This new and very simple arrangement consists of a cambered plate of best quality spring steel of rectangular shape, 3 mm. (1/8 inch) thick. The bolt hole is at one end out of centre with reference to the camber of the plate (fig. 1).

The length of the spring plate depends upon the fish plate and whether it is desired to use :

1. A plate for each bolt (fig. 2);
2. A plate securing two adjacent bolts against slacking back of the nuts (fig. 3);
3. A plate to the clip bolts holding rails to steel sleepers (fig. 4).

When tightening up a nut on an « S spring plate » an elastic reaction is produced on the face of this nut tending to cant the nut, which considerably increases the binding effect on the threads thereby locking the nut.

Figures 2, 3 and 4 show that the flattening of the end of the plate under the nut appreciably reduces the camber of the free portion, without however affecting the elasticity or the reaction caused by the plate, which remains constant and permanent.

In the application shown on figure 3, the length of the plate, measured along the curvature, being suitably chosen, the free end of the plate butts up to one of the flats of the adjacent nut, that is, it is within a distance less than the difference between the distance across the flats and the distance across the corners. This end therefore provides a stop 3 mm. (1/8 inch) in height against any slacking back of the nut on the bolt.

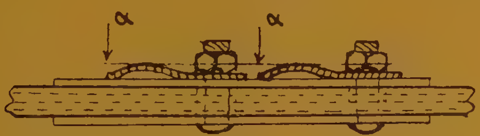
It may be also mentioned that the free end of the spring plate presses with some force upon the fish plate or other surface, since,

on account of the elastic action, the plate tends to resume its cambered form in the direction of the arrow (α). Consequently the plate absorbs nearly all the vibrations which occur in the joint under the action of passing trains, the direct transmission of which to the nut is undoubtedly one of the many causes of fish plate nuts slacking back.

It may be mentioned, however, that the « S spring plate » has the advantage over other devices of not preventing the free expansion and contraction of the rails at the joints.

Actual experiments have, on the other hand, shown that the nuts remain locked in a very satisfactory way for periods ranging from nine months to two years with the « S spring plate », the application of which, both on new lines and on old worn track, has maintained perfect tightness at the joints.

Fig. 2.



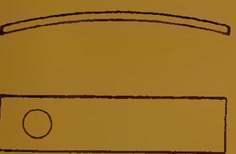


Fig. 1.

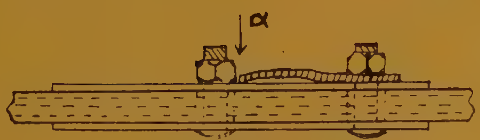


Fig. 3.

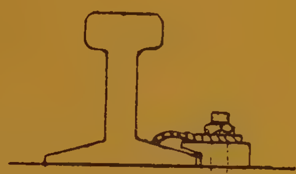


Fig. 4.

The invention and application of the « S spring plate » is based on the principle that the best method of preserving the tightness of a bolt for the longest possible time, especially in the case of rail fastenings, is to start with the nuts as tight as possible and to place under the nuts an elastic device which will retain its elasticity and act in such a way that the nuts cannot slack back on the thread under the effects of the vibrations to which the joint is subjected, so that they will remain tight almost indefinitely.

In this way, the effects of movement and shock, which result from the least slacking back of the nuts, are prevented, and this assures a very long life for the fish plates, clips, bolts and nuts.

It may be mentioned that the « S spring plate » cannot injure the thread of the bolt or the nut, or any of the parts of the joint, which therefore may be used again.

The quality of steel out of which these plates are made precludes any danger of breakage, either during or after tightening up.

The fact that it retains its elasticity assures its continued efficacy.

To conclude, we will quote a paragraph from *Traité des Chemins de fer* by Messrs. Flamache and Huberti, of the Belgian State Railways, the value of whose opinions is well recognised :

« Experience shows that nuts have a marked « tendency to slack back, and that to keep « these tight necessitates continued care and « inspection by the Permanent Way Depart- « ment. »

Tests which are now in hand show that the « S spring plate » successfully opposes the slacking back of nuts and maintains the tightness of the joint.

Its general use will therefore result in considerable economy, both in the upkeep and inspection of railways.

[625 .142.4 & 625 .143.5]

3. — Rail fastening for use with concrete sleepers.

Figs. 5 and 6, p. 171.

(*Les Chemins de fer et les Tramways.*)

The present tendency on railways is to replace creosoted or steel sleepers, with their many disadvantages, by concrete sleepers, which are unaffected by weather conditions, are easily packed, retain their shape, are quieter and are much cheaper.

So far no simple method of fastening the rail to the sleeper by which the rail can be easily fastened down or removed, whilst at the same time ensuring a reasonable durability of the concrete in contact with the sole plate, has been devised. The problem has not been solved satisfactorily by any of the many devices tested, including holding down castings embedded in the concrete when casting the sleeper, the castings having a tapped hole to suit the thread of coach screws.

This arrangement has many disadvantages. In the first place it is very difficult to obtain a proper bearing between the steel casting and the coach screw for the fastening. The rail consequently is not properly fastened down, the head of the coach screws have to be re-tightened frequently, and vibrations are set up which end by causing the concrete of the sleeper to disintegrate, thereby shortening the life of the sleeper and making the rail fastening less effective. Moreover, steel surfaces in contact rust up sooner or later, and make it difficult after a time to remove the coach screw when for any reason this is necessary. On the other hand, the holding down sleeves will not take every commercial coach screw, but only those which are accurately tapped to suit the holes. Finally, the gauge of the track is determined by the position of these sleeves and cannot be altered once the sleeper has been made.

The method of fixing rails to concrete sleepers described below, introduced by the Gennevilliers Steel Works, is cheap and simple, allows the rails to be removed easily, does not damage the sleepers, and is effective.

It consists essentially of a hollow steel or iron casting of special shape, with a rectangular slot at the top. This slot takes the T shaped head of a bolt, called the « foundation bolt », of the pattern used for fastening rails to the floor members of bridges or to steel sleepers.

The shank of this bolt also passes through a clip which holds the rail to the sleeper.

It is obvious that the bolt has only to be turned through 90°, that is to say, so that its rectangular head lies across the slotted hole in the casting, to ensure, on tightening up the nuts, that the rail is perfectly secured to the sleeper through the clip.

The single casting consists of a flange *a* and a hollow coned body *b*, so designed as to ensure that it will be completely surrounded by the concrete and the whole unit rigidly fixed in the sleeper. The upper hollow end of the casting is enlarged at *c*, and has cut in it the rectangular slot *d* through which the head of the holding down bolt can be passed; the casting is reduced at section *e* to prevent the bolt from dropping down inside during assembling.

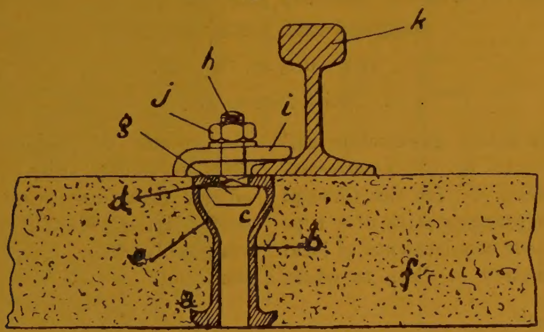
It will be seen in figure 5 that the casting *a* is embedded in the concrete sleeper *f*. The rectangular head *g* of the bolt *b* is passed through the slot. The bolt is then turned through 90° so that the head lies across the slot. The bolt is held in place by the casting; the clip *i* is put over it and is pulled down on to the flange of the rail by screwing up the nut *j*.

Another way of obtaining the same results is to use two sleeves as shewn in figure 6, not made up, as before, of single castings embedded in the concrete, but of two sleeves A and A′ side by side, connected by a small hollowed-out bar *l* cast in one with them to make the whole fitting sufficiently strong and rigid.

This arrangement has the following advant-

ages over the preceding type : it is more easily and accurately placed in the concrete when the sleeper is being made owing to the shape of the casting. It stops the hammering of the rail on the concrete of the sleeper, as the rail rests on the metal bar *l* connecting the two sleeves, and not directly upon the concrete. It also allows the gauge to be altered whilst using a single pattern of sleeper, by means of the transverse adjustment

Fig. 5.



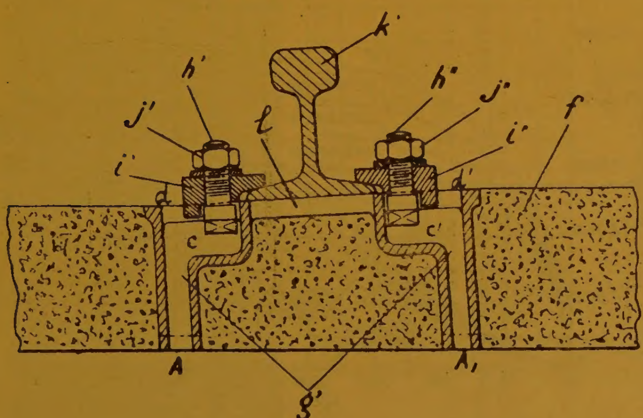


Fig. 6.

Figs. 5 and 6.

which can be obtained by moving the two clips which hold the rail. Further, the double sleeve can be given any desired inclination (one-twentieth, for example) towards the vertical centre line of the track, which allows the use of flat instead of tapered sole plates where these are employed.

The tops of the sleeves c' and c'' are of a different shape to that shewn in figure 5. Their slots d' and d'' are longer so that the bolt can be moved as required to ensure the fastening being properly adjusted when the clips have been moved.

i' and i'' are the rail holding down clips, h' and h'' the holding down bolts with their nuts j' and j'', k' the rail and f' the concrete sleeper.

In addition to the advantages already mentioned, as this method of fastening down the rails is similar to that commonly used with

steel sleepers or on floor members of bridges, no new types of bolts or clips are required, which makes the system still cheaper.

The general shape and material of the holding down casting (a sleeve either single or double) may be varied to suit requirements.

[**621** .335 (.73) & **621** .4 (.73)]

4. — Chicago & North Western Railway gas-electric rail motor car for passenger, baggage and postal services.

Figs. 7 to 9, pp. 172 to 173.

(From *Railway Age.*)

Three 72-foot combination gas-electric rail motor cars recently built by the Electro-Motive Company, Cleveland, O., for the Chicago & North Western are representative of a type designed to fulfil all the requirements of local steam passenger operation in districts where the volume of passenger traffic is limited. Cars previously built have either been designed for maximum passenger seating capacity or for limited seating capacity combined with reasonably large baggage and express compartments.

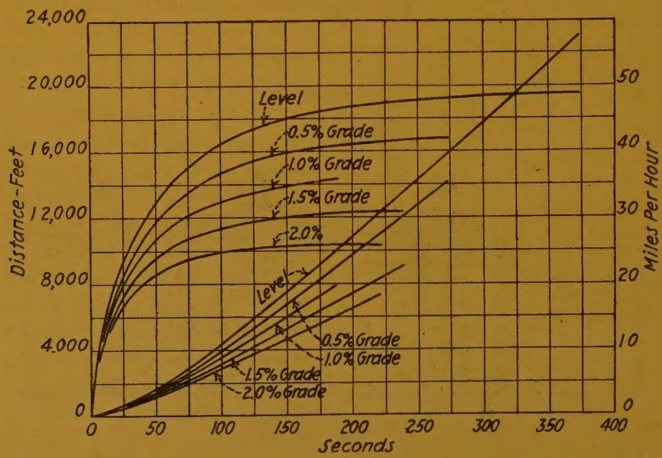


Fig. 7. — Speed-time and distance-time curves for the 72-foot motor car on varying grades.

These cars, it is expected, will be used in service on the Chicago & North Western between Harvard, Wis., and Kenosha, a distance of 44.3 miles, Clinton, Ia., and Anamosa, a distance of 71.4 miles, and Velle-Plaine, Ia., and Arkel, a distance of 63 miles. The steam train service between these points has been of the typical local passenger type, the make-up of the trains usually being a locomotive, combination mail and baggage car and one day coach weighing about 50 tons.

Quite often the demands of service are such that a relatively small number of passengers must be handled in conjunction with a normal amount of mail, express or baggage. The smaller, or 60-foot, motor car would in many cases necessitate the hauling of a trailer coach because of limited seating capacity and yet the traffic demand would hardly be enough to warrant the operation of the car and the trailer on certain schedules.

With the idea of meeting such conditions

Fig. 8. — 72-foot combination motor car built for the Chicago & North Western by the Electro-Motive Company.

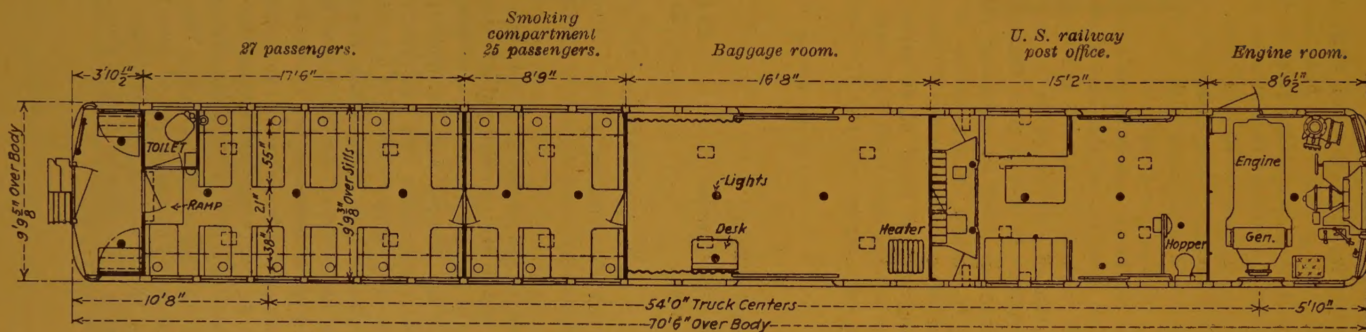


Fig. 9. — Floor plan of the 72-foot combination car built for the Chicago & North Western.

the Electro-Motive Company has built this type of car with a 70-ft. 6-in. body having ample baggage space, a smoking room seating 15 persons and a passenger compartment seating 27 persons, a total of 42 passengers. In addition a 15-foot mail compartment is provided, equipped in accordance with the requirements of railway postal service.

The table of average speeds shows a comparison of the operation of this car alone on level track and on a 0.5 % compensated grade with that of a 60-foot motor car hauling a 35-ton trailer. It will be seen that considerable advantage from an operating standpoint may be gained. A comparison of the acceleration rates show that the 72-foot car can be brought to a speed of 40 miles per hour in 90 seconds and 3 800 feet while the 60-foot car with trailer requires 170 seconds and 7 000 feet. Each has a braking deceleration rate of 1 1/2 miles per hour per second. Maximum speeds of approximately 50 miles per hour are possible.

Mechanical and electrical features.

These cars are typical of the single-unit gas electric cars built by this company which have been described in previous articles in the *Railway Age*. The power is generated by a single six-cylinder 200 H. P. Winton gasoline engine direct-connected to a 700-volt General Electric direct-current generator. Single-end controls are provided for two motors geared to the axles of the leading truck. The engines are designed for a normal operating speed of 1 000 revolutions per minute. Westinghouse type AML air brakes, with automatic features, are used.

Average speeds in miles per hour, level track.

Make-up of train :	Distance between stations (miles).									
	1	2	3	4	5	6	7	8	9	10
60-foot car and 35-ton trailer	24.0	30.0	33.5	36.0	37.0	38.5	39.5	40.0	40.5	41.0
72-foot car alone	28.0	34.0	37.5	39.5	41.0	42.5	43.0	43.5	44.2	44.8

Average speeds in miles per hour : grade 0.5 %, compensated.

60-foot car and 35-ton trailer. . . .	22.0	26.5	28.5	30.5	31.0	32.0	32.5	33.0	33.5	33.5
72-foot car alone	26.0	31.5	34.2	35.7	37.0	37.5	38.5	39.0	39.5	39.8

The total weight of the car is 90 000 lb.; the total length over coupler pulling faces is 72 ft. 9 5/16 in., and the width over eaves is 9 ft. 10 3/4 in. The car body and underframe are of steel construction and the interior is divided into five compartments and a vestibule platform. The lengths of the several compartments are as follows : engine room, 8 ft. 6 1/2 in.; mail compartment, 15 ft. 2 in.; baggage compartment, 16 ft. 8 in.; smoker, 8 ft. 9 in.; passenger compartment 17 ft. 6 in., and rear vestibule 3 ft. 10 1/2 in. The total length of the body is 70 ft. 6 in. The distance between truck centers is 54 feet with 6 ft. 6 in. wheel centers and 33 inches diameter wheels. The extreme height above the rail is 13 ft. 9 7/8 in.

As compared with the 60-foot car and standard trailer which together weigh approximately 75 tons, this 72-foot car, with an approximate weight of 45 tons, should comprise a relatively economical operating unit — the builders estimating that the over-all operating cost of the single larger car should be from 10 to 15 % less than that of one of the 60-foot cars handling a trailer. When operating conditions demand added capacity there is still reserve power enough in this car to handle one of the standard 35-ton trailers at a somewhat lower average speed, so it is believed that this type of car may be adapted to a great variety of operating conditions in local passenger service on branch lines.